

THE DESIGN OF A COMBINED  
STRESS FATIGUE JIG

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ALLEN JOHNSTON GILMORE

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COMBINED STRESS FATIGUE JIG

A. J. Gilmore

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THE DESIGN OF A  
COLD SURFACE HEATING SYSTEM

A. A. ULBRICH

Submitted in partial fulfillment of the requirements for the degree of  
Master of Science in Mechanical Engineering  
The University of Michigan  
Ann Arbor, Michigan  
1954



THE DESIGN OF A COMBINED  
STRESS FATIGUE JIG

by

Allen Johnston Gilmore  
Lieutenant, United States Navy

Submitted in partial fulfillment  
of the requirements  
for the degree of  
MASTER OF SCIENCE

United States Naval Postgraduate School  
Monterey, California  
1953

UNITED STATES OF AMERICA

DEPARTMENT OF COMMERCE

OF

NAVIGATION AND MARINE ENGINEERING  
AND MECHANICAL ENGINEERING

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## PREFACE

A search of the literature disclosed the limited amount of test data in the field of combined stress fatigue. This is probably due to the difficulties in production and operation of a suitable combined Stress Fatigue Machine. There are in existence several theories of failure under combined stresses, but not enough data for intelligent design criteria.

Between September 1952 and May 1953 the author designed and built a combined Stress Fatigue Jig to be used in conjunction with the Sonntag Universal Fatigue Testing Machine. This work was carried out at the United States Naval Postgraduate School, Monterey, California.

## PREFACE

A search of the literature disclosed the limited amount of test data in the field of combined stress fatigue. This is possibly due to the difficulties in production and operation of a suitable combined stress fatigue machine. There are in existence several machines of failure under combined stresses, but not enough data for intelligent design criteria.

Between September 1935 and May 1937 the author designed and built a combined stress fatigue rig to be used in conjunction with the General Universal Fatigue Testing Machine. This work was carried out at the United States Naval Postgraduate School, Monterey, California.

## TABLE OF CONTENTS

	Page
INTRODUCTION	1
FATIGUE TESTING IN GENERAL	2
PREVIOUS WORK IN FIELD OF COMBINED STRESS	2
DESIRED FEATURES IN NEW TESTING MACHINE	4
THEORY OF SONNTAG UNIVERSAL FATIGUE TESTING MACHINE	5
EFFECT OF DAMPING	8
DETAILS OF JIG DESIGN	12
THEORETICAL CHECK ON SONNTAG REQUIREMENTS	18
ACTUAL CHECK OF OPERATING JIG	24
BIBLIOGRAPHY	26
APPENDIX - TRANSIENTS	28
GRAPH OF DAMPING CHARACTERISTICS	29
GRAPH OF STRESSES VS ARM ANGLE	30
PARTS LIST	36
PRINTS OF JIG	38



1 INTRODUCTION  
2  
3  
4  
5  
6  
7  
8  
9  
10  
11  
12  
13  
14  
15  
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# TABLE OF SYMBOLS

$B$	Viscous damping coefficient
$E$	Young's modulus
$G$	Shear modulus
$I, I_z$	Moment of inertia
$K, K_s, K_c$	Spring constants
$l$	Length
$M$	Mass, Bending moment
$M_t$	Moment of torque
$P$	Force
$P_o$	Peak force
$x$	Displacement
$x_o$	Peak displacement
$\theta$	Angle
$\gamma$	Angle
$\omega$	Angular velocity
$\sigma$	Tensile stress
$\tau$	Shear stress
$\delta$	Deflection

# TABLE OF SYMBOLS

Viscous damping coefficient	$\eta$
Young's modulus	$E$
Mass moment of inertia	$I$
Stiffness of joints	$k$
Stiffness of springs	$k_s$
Length	$l$
Mass, bending moment	$M$
Moment of torque	$M_t$
Force	$F$
Peak force	$F_p$
Displacement	$x$
Static displacement	$x_s$
Angle	$\theta$
Angle	$\gamma$
Angular velocity	$\omega$
Tensile stress	$\sigma$
Shear stress	$\tau$
Deflection	$\delta$

## LIST OF ILLUSTRATIONS

	Page
Schematic Diagram of Gough's Fatigue Machine	2
Schematic Diagram of Sauer's Fatigue Machine	3
Sonntag Universal Fatigue Testing Machine	5
Schematic Diagram of Inertia Force Compensator	7
Schematic Diagram of System with Damping	9
Schematic Diagram of Fatigue Jig Loading	13
Schematic Diagram of Fatigue Jig	14
Distortion of Specimen	18
Schematic Diagram of Jig	21
Graph of Damping Characteristic	29
Graph of Predicted Stresses	30
Photograph of Jig	31
Photograph of Jig showing Instrumentation	32

# LIST OF ILLUSTRATIONS

Page	
2	Diagram of the Human Nervous System
3	Diagram of the Human Nervous System
4	Diagram of the Human Nervous System
5	Diagram of the Human Nervous System
6	Diagram of the Human Nervous System
7	Diagram of the Human Nervous System
8	Diagram of the Human Nervous System
9	Diagram of the Human Nervous System
10	Diagram of the Human Nervous System
11	Diagram of the Human Nervous System
12	Diagram of the Human Nervous System
13	Diagram of the Human Nervous System
14	Diagram of the Human Nervous System
15	Diagram of the Human Nervous System
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19	Diagram of the Human Nervous System
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24	Diagram of the Human Nervous System
25	Diagram of the Human Nervous System
26	Diagram of the Human Nervous System
27	Diagram of the Human Nervous System
28	Diagram of the Human Nervous System
29	Diagram of the Human Nervous System
30	Diagram of the Human Nervous System
31	Diagram of the Human Nervous System
32	Diagram of the Human Nervous System

## SUMMARY

The objective of this project was the design and production of a combined Stress Fatigue Machine for use at the U. S. Naval Postgraduate School. The design was carried out using the Sonntag Universal Fatigue Testing machine as the basic component and then producing a mechanism which would exert the desired stresses in a specimen.

Tests of the completed jig indicate that the assembled machine does function as predicted. The results of these tests are included as an appendix to this thesis. The errors indicated are attributable to the testing technique. It is believed that further calibration and testing will be able to reduce these errors greatly.



## ABSTRACT

The objective of this project was the design and construction of a mechanical device which would serve as a means of measuring the force exerted by the human hand. The design was carried out using the mechanical design process as the basic component and then producing a mechanical device which would serve the desired purpose in a practical manner.

Tests of the completed device were conducted using a standard test method. The results of these tests are contained in an appendix to this report. The errors indicated are within the limits of the testing technique. It is believed that further refinement and testing will be able to reduce these errors greatly.

## INTRODUCTION

In an effort to increase the efficiencies of mechanical apparatus, the trend has been continually to reduce the size and weight of machine parts, and to increase speeds. With the evolution of smaller and higher speed machinery, the phenomenon of fatigue has become an ever increasingly important limitation upon design. In the last few decades, much information has been gained by extensive research in the field of simple fatigue, and only recently has there been any research in the field of fatigue under conditions of multiple dynamic stresses. This is a practical trend as most machine parts are subject to more than one dynamic stress. In pursuit of this trend, the project of the design of equipment to run fatigue tests under multiple dynamic stresses was undertaken. Once having set upon this problem, and analyzing the methods of other people in the field, the project narrowed down to the design of a special jig for the Sonntag SF-1-U Universal Fatigue Testing Machine, which would produce simultaneous dynamic bending and torsion in a specimen.

## INTRODUCTION

In an effort to increase the efficiency of mechanical systems, the trend has been steadily to reduce the size and weight of machine parts, and to increase speed. With the evolution of smaller and higher speed machinery, the phenomenon of fatigue has become an ever increasingly important consideration upon design. In the last few decades, much information has been gained by extensive research in the field of fatigue testing, and only recently has it been applied to the field of design under conditions of variable dynamic stresses. This is a practical trend in most machine parts and subject to more than one dynamic stress. In pursuit of this trend, the project of the design of equipment for the fatigue tests under variable dynamic stresses was undertaken. Close having set upon this project, and analyzing the nature of other people in the field, the project was narrowed down to the design of a special jig for the testing of U-shaped fatigue testing machine, which would produce simultaneous dynamic bending and torsion in a specimen.



## FATIGUE TESTING IN GENERAL

There are available and in use throughout the world, many different fatigue testing machines which will give good results for simple fatigue. These consist of: bending machines which use rotating and non-rotating specimens loaded either in pure bending or as a cantilever, tension-compression machines, and torsion machines. Through the years much valuable data has been obtained. However, this data cannot be transposed to the field of combined stresses with any degree of adequacy for practical machine design.

### PREVIOUS WORK IN FIELD OF COMBINED STRESS

Little data is available in the field of fatigue under combined stresses, due to the difficulty in producing and operating a combined fatigue machine. Gough (6) in England has done extensive work in this field using a resonant frequency machine acting on a specimen loaded as a cantilever.

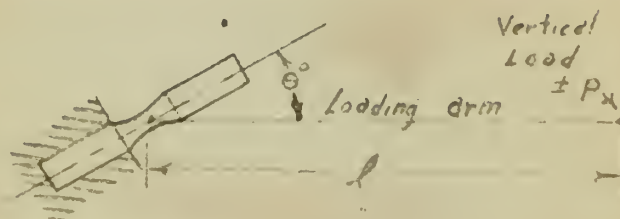


Figure 1

Schematic Diagram of Gough's Fatigue Machine

## INTRODUCTION

In an effort to increase the efficiency of mechanical systems, the trend has been continually to reduce the size and weight of machine parts, and to increase speed. With the evolution of smaller and higher speed machinery, the demands of design are becoming ever increasingly important. In the past, the designer, with information from tests, relied on extensive research in the field of static strength, and only recently has there been any progress in the field of dynamic strength. This is a potential source of great machine parts and subject to more than one dynamic stress. In periods of this trend, the project of the design of equipment for the fatigue tests under multiple dynamic stresses was undertaken. Once having set upon this project, and analyzing the methods of other people in the field, the project was narrowed down to the design of a special rig for the General Electric 21-1 Universal Fatigue Testing Machine, which would produce simultaneous dynamic bending and torsion in a specimen.

## FATIGUE TESTING IN GENERAL

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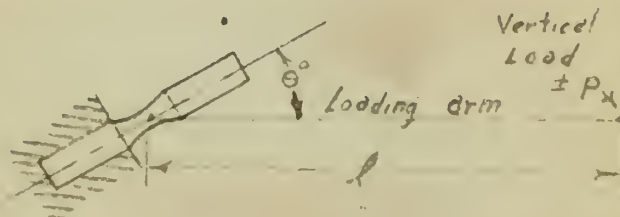


Figure 1

Schematic Diagram of Gough's Fatigue Machine



There are available and in use throughout the world, every different type of testing machine which will give good results for the purpose. These include all testing machines which are not too complicated and expensive to build, and which are not too complicated to use. However, this data is not to be compared to the data of standard tests of the type of machine used.

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There was a weakness in the field of letters under consideration, due to the difficulty in procuring and securing a suitable letter machine. Govt. (S) in England has been extensive work in this field using a permanent temporary machine being on a specimen loaded to a satisfactory.

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Approved: \_\_\_\_\_  
Special Agent in Charge

This machine uses a standard size specimen of non-constant cross section in the region under test, in order to predict with reasonable accuracy the stresses in the specimen at the point of failure. Sauer (14) did a limited amount of work by using the standard torsion jig of the Sonntag Universal Fatigue Testing Machine, and by removing the bearing adjacent to the torque arm. This machine again used a specimen of non-constant cross section loaded as a cantilever. In each

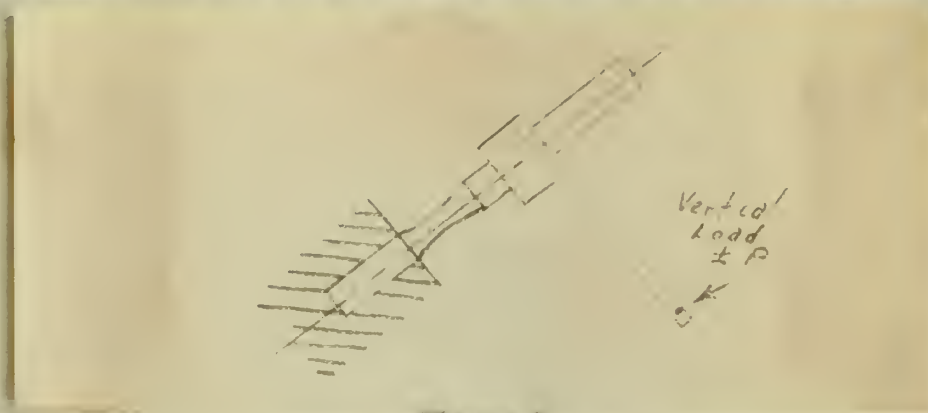


Figure 2

#### Schematic Diagram of Sauer's Fatigue Machine

case, the cantilever loading of the specimen necessitates a necked down section of the specimen in order to predict the point of failure and the stresses existing at that point. However, as the exact point of failure is a statistical function, not always occurring at the section of minimum cross section, exact determination of the stresses at the point of initial failure is impossible. Gough with a specially built machine and with a careful choice of specimen shape and size has produced excellent reproducible results, which are universally accepted. Sauer's results are of doubtful value because of the limited scope of test, and his failure to show adequately that the induced stresses were

This machine was a standard after specimen of two-dimensional  
 motion in the region under test. It was in fact, with possible  
 accuracy the distance in the specimen at the point of failure. But  
 (1) did a standard amount of work up until the standard testing life of  
 the bearing between fatigue testing machine, and by testing the  
 bearing subject to the fatigue test. This machine also used a spec-  
 imen of two-dimensional stress which looked as a cylinder. In each



Figure 1

### Standard Machine of Standard Fatigue Machine

Now, the machine testing of the specimen necessitates a method  
 that section of the specimen in order to provide the point of failure  
 and the specimen existing at that point. However, as the work point  
 of failure is a statistical function, not always occurring at the sec-  
 tion of minimum cross section, some consideration of the stresses at  
 the point of initial failure is imperative. Such is a specialty  
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 Some's results are of limited value because of the limited scope of  
 test, but the theory is more completely that the limited stresses were



as specified. In short, his jig had two degrees of freedom while the theory only considers one. It is perhaps significant that a large percentage of his specimens failed in the grips.

#### DESIRED FEATURES IN NEW TESTING MACHINE

In the design of a combined fatigue machine, there are many different conditions which must be satisfied. First it is desirable to use as many components "on hand" as possible from a standpoint of simplicity and economy. Secondly, it is desirable to have the specimen subjected to a dynamic force of constant amplitude rather than a constant amplitude of motion. This is desirable, because any change in physical properties or yield in a specimen of the constant amplitude of oscillation type specimen during a test will result in a test of unknown parameters. This leads directly to the desirability of using the Sonntag SF-1-U Universal Fatigue Testing Machine. This machine is capable of exerting a sinusoidal dynamic force varying in amplitude from zero to 1000 pounds, regardless of the rigidity of the test specimen. In addition this machine may be given a preset static force of from zero to 1000 pounds. Therefore the maximum setting would be sinusoidal force with peaks values of zero and 2000 pounds. In addition, the large table top of this machine with provision for fastening down various jigs provides a very flexible system for utilizing its full capabilities.

It is considered desirable to produce the tension in the specimen





by placing the test section in pure bending, to eliminate some of the undesirable features of previous machines which load the specimen as a cantilever. In addition the jig should be so designed that the stresses in specimen may be varied from pure tension (bending) to pure torsion so any desired ratio may be obtained. In view of the requirements the Sonntag Universal Fatigue Testing machine was chosen as the basic component of the combined fatigue testing machine.

#### THEORY OF SONNTAG UNIVERSAL FATIGUE TESTING MACHINE

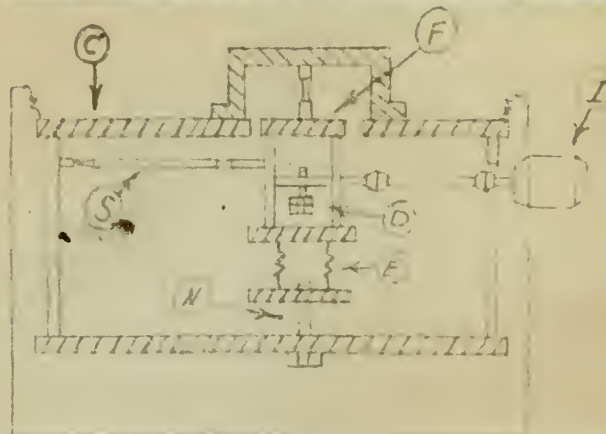


Figure 3

#### Sonntag Universal Fatigue Testing Machine

The function of the Sonntag Universal Fatigue Machine is to apply a vertical vibratory force to any specimen attached between the heavy stationary frame (C) and the reciprocating platen (F). This force can have any static component from zero to 1000 pounds either in tension or compression and any alternating component from zero to 1000 pounds. When operating in tension or compression alone, it is possible to have a maximum vibrating load fluctuating from zero to 2000 pounds.

by giving the last member in our building, an illustration, some of the  
unusually low level of growth which has been observed in  
a number of cases. In addition, the last member is so defined that the  
member in question may be taken from the building (building) to give  
formation as an object which may be obtained. In view of the regular  
rents the building owner is not to be taken into account as the  
basic component of the building. The building is a building.

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The position of the United Nations Relief Commission is to apply a critical viewpoint to any question arising between the two elements (5) and the Commission (6). This force can have any effect compared from 1000 pounds to 1000 pounds, compression and any kind of compression, from 1000 pounds, when operating in terms of compression alone, it is possible to have a certain amount of compression from 1000 pounds.

The alternating force is produced by the unbalanced rotating mass (D) which is supported in the oscillating frame by two bearings. This mass is rotated through flexible couplings by an 1800 RPM synchronous motor (I). The amount of dynamic force exerted on the platen is set by adjusting the amount of eccentricity of the mass (D). Only the vertical component of this unbalanced force is exerted upon the specimen. The horizontal components are absorbed by guides with elastic hinges (S).

The Compensator springs (E) have one end attached to the oscillating frame and the other end attached to a plate which is in turn solidly fixed to the stationary frame. The purpose of these springs is to absorb all unknown inertia forces produced by the reciprocating masses and prevent these inertia forces from reaching the specimen. In order to function properly, the rigidity of the compensating springs is such that the natural frequency of the spring together with the reciprocating mass without specimen is the same as the operating frequency, 1800 cycles per minute. To accomplish this, provision is made for adjusting the mass to give resonance.

Preload is applied by adjusting the screw (N) which will place an initial tension or compression on the compensating spring. In addition there are micro switches to shut off the motor when the specimen fails and a counter to record the number of stress cycles to failure.



The compensating force is provided by one or more compensating springs (1) which is supported in one or more frames (2) and is connected to the frame (1) by one or more springs (3). The amount of spring force exerted on the frame is set by adjusting the amount of extension of the spring (1). Only the vertical component of the spring force is exerted upon the specimen. The horizontal components are absorbed by guides with sliding rings (4).

The compensating springs (1) have one end attached to the casing of the frame and the other end attached to a plate which is in turn rigidly fixed to the stationary frame. The purpose of these springs is to absorb all unknown inertia forces produced by the accelerating masses and prevent these inertia forces from reaching the specimen. In order to function properly, the rigidity of the compensating springs is such that the natural frequency of the spring together with the specimen is well above the operating frequency. In order to achieve this, provision is made for adjusting the mass to give resonance.

Twisted is applied to adjust the mass (1) when the plate on which the specimen is supported on the compensating spring. In addition there are also adjusters to shift all the masses when the specimen is moved to a new position of the specimen of these adjusters to follow.

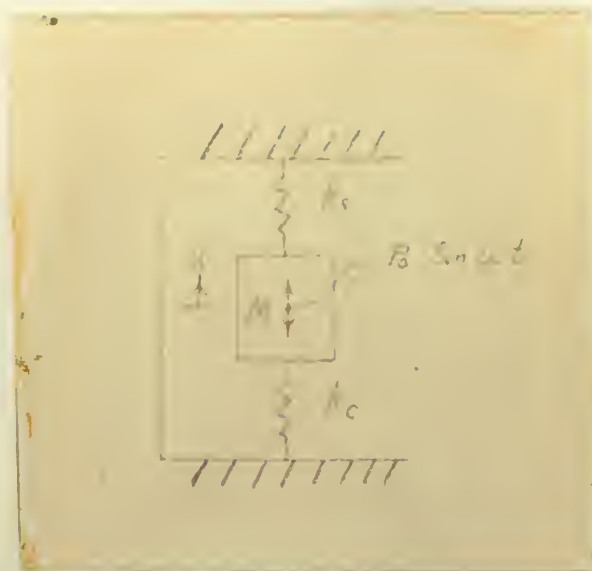


Figure 4

#### Schematic Drawing of Inertia Force Compensator

The schematic drawing of the testing machine as shown has the following equation of motion:

$$M \frac{\partial^2 x}{\partial t^2} + Kx = P_0 \sin \omega t$$

where  $K = K_s + K_c$

$K_s$  = spring constant of specimen

$K_c$  = spring constant of compensating springs

$x$  = displacement of Mass M

The steady state solution of this equation is:

$$x = x_0 \sin \omega t = \frac{P_0 \sin \omega t}{K - M\omega^2}$$

Considering only peak values and substituting

$K_s + K_c$  for  $K$  we have:

$$x_0 = \frac{P_0'}{(K_c + K_s) - M\omega^2}$$

Figure 1

Generalized Theory of Linear Three-Parameter

The generalization of the linear model is shown in the

following equation of motion:

$$M \frac{d^2 x}{dt^2} + Kx = F_0 \sin \omega t$$

where:  $F_0 = F_0 \sin \omega t$

$F_0$  = spring constant of specimen

$F_0$  = spring constant of supporting spring

$K$  = displacement of mass  $M$

The steady state solution of this equation is:

$$x = \frac{F_0 \sin \omega t}{K - M \omega^2} = F_0 \sin \omega t \cdot X$$

Substituting only two values and calculating

$F_0 + F_0 \sin \omega t$

$$F_0 = \frac{F_0}{1 + \frac{M \omega^2}{K}}$$



however by design  $K_c = M \omega^2$

$$\text{Therefore } X_o = \frac{P_o}{K_s}$$

Thus the force exerted on the specimen is that produced by the oscillating force irrespective of specimen rigidity of the specimen changes.

The mass of the reciprocating system was adjusted to give resonance without the specimen in place. The small additional rigidity of the specimen will raise the natural frequency of the system slightly. This is desirable to insure that the force and displacement are always in the proper phase.

The machine is driven by a synchronous motor to insure that the frequency of the forcing function is constant throughout the test. As the rigidity of the springs is fixed in manufacture some means of tuning the system must be provided. This is accomplished by adding tuning weights to the reciprocating frame such that the total mass, without specimen will place the system in resonance.

#### EFFECTS OF DAMPING

This analysis of the Universal Fatigue Testing Machine assumes that the vibrating system possesses zero damping. In his analysis of the machine B. J. Lazan (9) says, "In practically every case the damping in the system is sufficiently low to make the resulting errors

however by taking  $\lambda' = \frac{1}{2} \lambda$

$$\text{Therefore } \lambda' = \frac{\lambda}{2}$$

Thus the force exerted on the specimen is that produced by the  
restoring force irrespective of specimen rigidity or the specimen  
geometry.

The mass of the restituted specimen was referred to the rest-  
mass of the specimen in place. The small additional rigidity  
of the specimen will raise the natural frequency of the system slightly.  
This is desirable inasmuch that the force and displacement are always  
in the proper phase.

The system is driven up a resonance which to insure that the  
frequency of the driving function is constant throughout the test. In  
the vicinity of the system is fixed in resonance with mass of  
spring the system will be provided. This is accomplished by adding  
weight which is the restituted mass and that the total mass,  
without specimen will place the system in resonance.

#### SYSTEM OF TESTING

This system of the universal testing machine system  
that the vibrating system possesses very simple. In its subject of  
the machine is a mass (?) force, is practically every time the damp-  
ing in the system is sufficiently low to make the resulting error



negligible". However the internal damping of all parts and particularly the inherent non-linear damping characteristics of anti-friction bearings cannot be so lightly ignored.

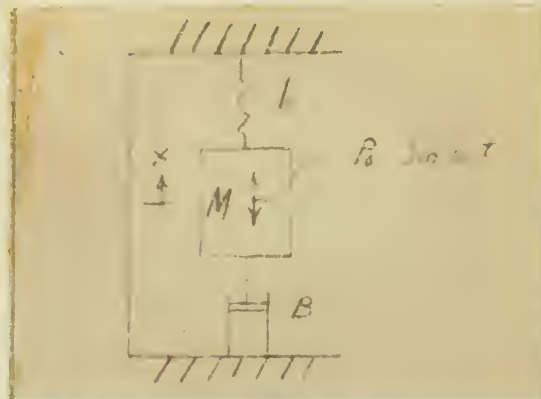


Figure 5

#### Schematic of System with Damping

If we consider all damping to be external viscous damping and analyze the system thoroughly, then the relative significance of this damping can be determined.

The equation of motion of this system is:

$$M \frac{d^2x}{dt^2} + B \frac{dx}{dt} + Kx = P_0 \sin \omega t.$$

The steady state solution of this is:

$$x = \frac{P_0 \sin(\omega t - \phi)}{\sqrt{(K - M\omega^2)^2 + \omega^2 B^2}}$$

$$\text{where } \phi = \tan^{-1} \frac{B\omega}{K - M\omega^2}$$

substituting  $K_s + K_c$  for  $K$  and  $K_c = M\omega^2$

$$x = \frac{P_0 \sin(\omega t - \phi)}{\sqrt{K_s^2 + \omega^2 B^2}}$$

$$\phi = \tan^{-1} \frac{B\omega}{K_s}$$

...the ... of the ...  
 ...the ... of the ...  
 ...the ... of the ...



...the ... of the ...  
 ...the ... of the ...  
 ...the ... of the ...

$$M \frac{dx}{dt} + N \frac{dy}{dt} = P(x, y)$$

The exactness condition is

$$\frac{\partial M}{\partial y} = \frac{\partial N}{\partial x}$$

...the ... of the ...

$$\frac{\partial M}{\partial y} = \frac{\partial N}{\partial x}$$

From this it may be seen that the amplitude and hence the stresses in the specimen are no longer directly a function of the rigidity of the specimen and the forcing function. The significance of the error thus introduced into the system must be checked by some other means.

A check was made of the damping characteristic of the machine in three conditions: the machine alone, with the 15 1/4" torsion jig installed without a specimen, and the 15 1/4" torsion jig with an aluminum specimen. This was accomplished by placing an SR-4 strain gage on one of the elastic hinges of the machine and using the signal of this gage to operate a brush recorder. Then by causing the system to oscillate by compressing the compensating springs and releasing, the following damping coefficient were determined.

Machine alone	$K = 0.05$
Torsion jig without specimen	$K = 0.125$
Torsion jig with aluminum specimen	$K = 0.130$

where amplitude  $X = X_0 e^{-Kt}$

To convert to the derived function B:

$$K = \frac{B}{2M \sqrt{\frac{KS}{M} - \frac{B^2}{4M^2}}}$$

Assuming the spring constant  $K_s \approx 10,000$  lb per in.

and the logarithmic decrement  $K \approx 0.1$

then by simplifying  $B^2 \approx \frac{25}{BK_s}$



From this it may be seen that the magnitude and shape of the response  
 to the specimen are no longer directly a function of the rigidity of  
 the specimen and the forcing function. The relationship of the error  
 thus introduced into the system may be checked by some other means.  
 A check was made of the damping characteristics of the machine in  
 three positions. The machine alone, with the 12 1/2 lb. weights 12 in.  
 spaced along a specimen, and the 25 1/2 lb. weights 12 in. in each  
 end specimen. This was accomplished by placing an 80-lb. weight on  
 one of the elastic hinges of the machine and noting the signal at this  
 rate to operate a brush recorder. Then by cutting the system so that  
 it is by compressing the supporting springs and releasing, the  
 following damping coefficients were determined.

Machine alone	$\gamma = 0.25$
Weights 12 in. apart	$\gamma = 0.18$
Weights 12 in. apart	$\gamma = 0.10$

$$\text{where } \gamma = \frac{1}{2} \ln \frac{1}{\rho} \quad \rho = \frac{x_1}{x_2}$$

To convert to the desired function:

$$\gamma = \frac{1}{2} \ln \frac{1}{\rho} = \frac{1}{2} \ln \frac{1}{\frac{1}{1 + \frac{2\pi^2 \eta}{\omega^2 m}}} = \frac{1}{2} \ln \left( 1 + \frac{2\pi^2 \eta}{\omega^2 m} \right)$$

Assuming the spring constant  $k = 10,000$  lb per in.

$$\text{and the logarithmic decrement } \delta = 0.1$$

$$\text{then by substituting } \delta = \frac{1}{2} \ln \frac{1}{\rho}$$

$$\frac{0.1}{2} = \frac{1}{2} \ln \left( 1 + \frac{2\pi^2 \eta}{\omega^2 m} \right)$$



Then the expression  $\sqrt{K a^2 + \omega^2 B^2}$  will approach  $K_s$  to within one part in  $10^4$  and can be considered equal to  $K_s$  with negligible error. Thus the damping in the standard Sonntage Fatigue Jig is considered negligible.

It is a reasonable assumption that the damping in the new jig will be of the same order of magnitude as in the existing machine and therefore be negligible. However, this will be thoroughly tested upon completion of the new jig.

The use of the Sonntag Universal Fatigue Testing Machine imposes several limitations. First the equivalent weight of the system must be within the limits of compensation of the machine (15.4 lbs.). Secondly the maximum amplitude of oscillation must not exceed 0.44 inches. Thirdly, the stress ranges desired in the specimen must be within the capability of the forces available.

The first question is whether the error in the  
 in 10 and not as considered equal to 10 with negligible error. Thus the  
 due to the constant  $\frac{1}{2} \pi$  is considered negligible.  
 It is a reasonable assumption that the change in the new  $\frac{1}{2} \pi$  will  
 be of the same order of magnitude as in the existing machine and there-  
 fore negligible. However, this will be thoroughly tested upon com-  
 parison of the new  $\frac{1}{2} \pi$ .  
 The use of the constant  $\frac{1}{2} \pi$  in the existing machine involves  
 several limitations. First the constant value of the error must  
 be within the limits of compensation of the machine (1.5 per cent). Sec-  
 ondly the machine requires at least 1000 cycles per second (1000 cps).  
 Thirdly, the error ranges desired in the machine must be within the  
 capability of the force available.

## DETAILS OF JIG DESIGN

The method chosen to accomplish the desired loading upon a specimen consists of mounting the specimen over the oscillating platen and supporting it with self aligning bearings which are attached to the platen. Attached to each end of the specimen is an arm which is capable of being swung in an arc to give any desired angle with the axis of the specimen. The ends of these arms are attached to the table top of the stationary frame.

A bending stress is imposed on the specimen by causing equal and opposite bending moments on the ends of the specimen. The moment is created by arms which have one end fastened to the fixed frame of the machine and one fixed to the oscillating platen. By symmetry the forces are exerted equally on the two lever systems so that the specimen is placed in pure bending. A torsional stress is imposed through the same system. If the levers lie perpendicular to the axis of the specimen and are placed symmetrically, then as a vertical force is exerted on the platen this will cause a torsional force to be exerted upon the specimen. If the lever arms are placed in a position between the axis of the specimen and perpendicular to the axis, then the combined stresses will be exerted on the specimen in a proportion which may be readily determined from the geometry of the system. These stresses will bear the same ratio for static as well as dynamic forces.

The bearing at the ends of the specimen must be capable of exerting a vertical force necessary to produce the required couples, but it also



## DETAILS OF THE METHOD

The method chosen to accomplish the desired loading upon a specimen

consists of mounting the specimen over the oscillating plates and supporting it with self-aligning bearings which are attached to the plates. Attached to each end of the specimen is an arm which is capable of being swung in or out to give any desired angle with the axis of the specimen. The ends of these arms are attached to the table top of the stationary frame.

A loading stress is imposed on the specimen by causing equal and opposite bending moments on the ends of the specimen. The moment is created by arms which have one end fastened to the fixed frame of the machine and one fixed to the oscillating plates. By opposing the forces are exerted equally on the two lower plates so that the specimen is placed in pure bending. A vertical stress is imposed through the arms. If the lower arm perpendicular to the axis of the specimen and are placed symmetrically, then a vertical force is exerted on the plates. This will cause a horizontal force to be exerted upon the specimen. If the lower arm are placed in a position between the axis of the specimen and perpendicular to the axis, then the combined stresses will be exerted on the specimen in a direction which may be readily determined from the geometry of the system. These stresses will bear the same ratio for static as well as dynamic forces.

The loading at the ends of the specimen must be capable of supplying a vertical force necessary to produce the required couples, but it is also

must permit the bending and torsion to be fully transmitted to the specimen. For this reason self aligning bearings are used at this point. As the stresses in the specimen are changed and it deflects there is a small change in the axial length of the specimen. To eliminate any unknown axial forces, one of the vertical members holding these self aligning bearings must be made free to travel along the axis, while still exerting its vertical force. This is accomplished by hinging one of the bearing holders at the platen so that this second order motion can take place freely. Similarly deflection of the specimen will result in the motion of the extremity of the lever along its own axis in addition to the small motion along the axis of the specimen. As it is desirable to exert only a vertical force, self aligning bearings are placed at both ends of each holding down link.

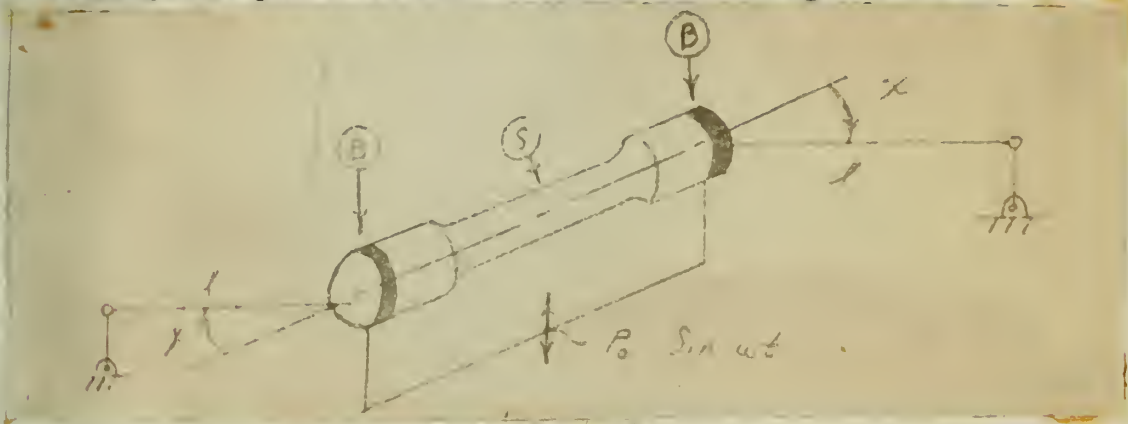


Figure 6

#### Schematic Drawing of Fatigue Jig Loading

The force  $P_0 \sin wt$  is exerted on the specimen (S) through two self aligning bearings (B) mounted on either end of the specimen. This force is resisted by the two symmetrically located torque arms.



must point the bearing and distance to the ship transmitted to the  
specimen. For this reason self-aligning bearings are used at the  
point. In the station in the specimen are changed and it follows  
there is a small change in the actual length of the specimen. The  
alignment of the specimen and the bearing, one of the vertical members hold-  
ing these self-aligning bearings must be made free to travel along the  
axis, while still exerting the vertical force. This is accomplished by  
sliding one of the bearing rollers at the place so that the second  
vertical member can move freely. Similarly, deviation of the spec-  
imen will result in the motion of the center of the lower along the  
own axis in addition to the small motion along the axis of the speci-  
men. As it is desirable to have a vertical force, self-aligning  
bearings are placed at both ends of each holding down plate.



Figure 6  
Schematic diagram of bearing alignment  
The force  $P$  is exerted on the specimen (2) through the  
self-aligning bearings (3) mounted on either end of the specimen. This  
force is resisted by the two vertically located support arms.

By virtue of the geometry of this loading, the bending moment in the specimen will be  $\frac{P_0}{2} \sin \omega t (l \cos \phi)$  and the moment of torque will be  $\frac{P_0}{2} \sin \omega t (l \sin \phi)$ . Thereby varying the angle  $\phi$  the desired proportions of bending moment and torque, both dynamic and static, from pure bending to pure torsion may be exerted on the specimen.

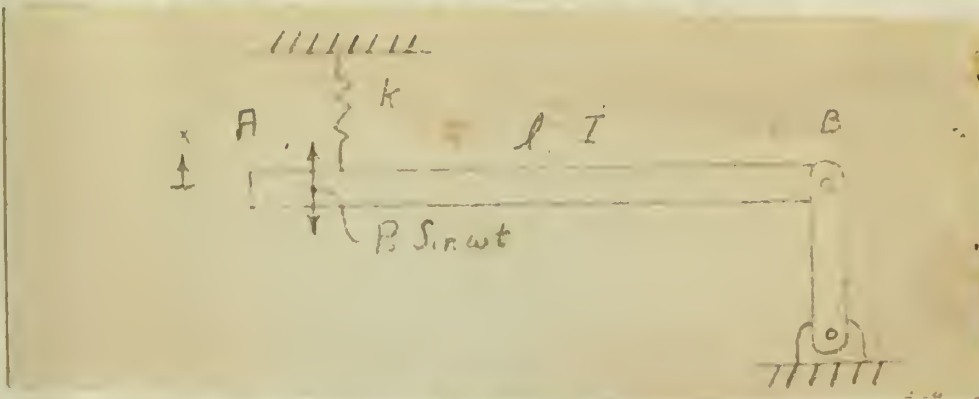


Figure 7

#### Schematic Drawing of Fatigue Jig

The configuration of the jig is such that the mass is not constrained so that the end A is forced to oscillate in a vertical path. Neglecting the small motion of the hinge B the equation of motion of this system is  $\frac{I}{l^2} \frac{d^2 x}{dt^2} + Kx = P_0 \sin \omega t$ .

The solution of this equation is:

$$x = x_0 \sin \omega t = \frac{P_0 \sin \omega t}{K - \frac{I}{l^2} \omega^2}$$

if  $\frac{I}{l^2} \omega^2 = Kc =$  a constant, which can be tuned for then

$$x = \frac{P_0 \sin \omega t}{Ks}$$

By virtue of the geometry of this triangle, the position vector to the  
 apex will be  $\frac{1}{2} \sin \theta$  and the vector to the base will be  $\frac{1}{2} \cos \theta$ .  
 The vector to the apex is  $\frac{1}{2} \sin \theta$ . The vector to the base is  $\frac{1}{2} \cos \theta$ .  
 The position of the apex and base, with respect to the origin, then  
 pure bending is pure rotation and is exerted on the specimen.



Figure 1

Geometric description of Figure 1

The deformation of the lig is such that the mass is not  
 stretched so that the end is forced to oscillate in a vertical path.  
 Neglecting the small motion of the lig the equation of motion of

$$\text{This system is } \frac{1}{2} \frac{d^2 x}{dt^2} + Kx = \frac{1}{2} \sin \omega t.$$

The solution of this equation is:

$$x = \frac{1}{2} \frac{\sin \omega t}{\omega^2 - K} + \frac{K}{\omega^2 - K} \sin \omega t$$

is  $\frac{1}{2} \frac{\sin \omega t}{\omega^2 - K} + \frac{K}{\omega^2 - K} \sin \omega t$  a constant, which can be added

$$x = \frac{1}{2} \frac{\sin \omega t}{\omega^2 - K} + \frac{K}{\omega^2 - K} \sin \omega t$$



or for peak deflections

$$P_o = K_s X_o$$

Therefore the inertia force compensator is still effective for this configuration. This system is not entirely accurate as there is a small axial motion of the hinges B. However, the error involved is negligible as will be shown later.

The specimen had to be checked so that the bending moments and torsional moments could be exerted upon it positively and with no lost motion. This was accomplished by having the moments exerted on the specimen holders. A tapered collar or collet fits over each end of the specimen and seats against a shoulder. The specimen is drawn into the specimen holders by a screw at each end which seats the tapered collar against a mating taper in the specimen holder. This allows the transmission of the moments to the specimen and eliminates the possibility of lost motion.

As shown in the description of the jig it is necessary to use the self aligning bearing in order to achieve the necessary degrees of freedom. The choice of these bearings was limited by the usual problems which are found in the use of antifriction bearing operating under reciprocating conditions. Any self aligning ball bearing when checked with the formula as recommended by Patterson (12) was either inadequate or too large. The only adequate bearing within the weight



of the basic definition

$$\frac{p}{n} = \frac{K}{n} \frac{1}{2}$$

Therefore the linear force component is still effective for this configuration. This system is not entirely accurate as there is a small initial action of the spring. However, the error involved is negligible as will be shown later.

The specimen had to be checked so that the loading moment and torsional moment could be entered upon it positively and with no loss of accuracy. This was accomplished by having the moment entered on the specimen holder. A tapered collar of mild steel was used on the specimen and seats against a shoulder. The specimen is drawn into the specimen holder by a screw at each end which seats the tapered collar against a mating taper in the specimen holder. This allows the transmission of the moment to the specimen and eliminates the possibility of lost motion.

As shown in the description of the test it is necessary to use the self-aligning bearing in order to achieve the necessary degree of freedom. The choice of these bearings was limited by the usual problems which are found in the use of antifriction bearings operating under nonreciprocating conditions. Any self-aligning ball bearing when checked with the formula recommended by Peterson (11) was of the type of two large. The only adequate bearing within the weight

limitations were aircraft type self aligning needle bearing. The choice of the bearing for the central bearing holder was simplified by the fact that similar bearings are used by Sonntag in their bending jig.

In order to meet the requirement that the jig be adjustable from pure bending to pure torsion, a false table top was necessary. This table top has circular slots to allow continuous adjustment.

Most oscillating components are manufactured from aluminum in an effort to remain within the weight limitations. However, parts such as the specimen holders and tapered collets are made of heat treated steel for strength.

The weak point in this jig is the strength of torque pins (Parts #3). Considering the force per inch of length (F) to be proportional to the distance from the center line of the specimen, and a case where the force exerted by the vertical link equal to five hundred pounds.

$$M = 2 \int_0^{1.75} 7 x^2 dx = 500 \times 6 \quad \text{lb in.}$$

$$f = 515 \frac{\text{lb}}{\text{in.}}$$

Therefore the moment at base of the cantilever pin is

$$M = \int_0^{0.75} 515(1+x) x dx = 214 \quad \text{lb in.}$$

$$\text{then } \sigma = \pm \frac{Mc}{I} Kc = \pm \frac{214 \times 32}{\pi \left(\frac{3}{8}\right)^3} \times 1.5 = \pm 62,500 \text{ PSI}$$

illustrated as were already type with alignment needed bearing. The choice of the bearing for the central bearing holder was simplified by the fact that similar bearings are used by companies in their bearings fig.

In order to meet the requirement that the fig. be adjustable from here bearing to here bearing, a false table top was necessary. This table top has circular slots to allow continuous adjustment. Most oscillating components are manufactured from aluminum in an effort to remain within the weight limitations. However, parts such as the specimen holders and tapered collars are made of heat treated steel for strength.

The weak point in this fig. is the strength of tapered glass (Peters 1961). Considering the force per inch of length (1/2) to be proportional to the distance from the center line of the specimen, and a curve where the force exerted by the vertical line equal to five hundred pounds,

$$M = 2 \int_0^{1.12} 7 \times 10^4 x = 200 \times 10 \times 10 \times 10$$

$$1 = \frac{212}{1.12} \times 10$$

Therefore the moment at base of the condenser arm is

$$M = \int_0^{0.12} 212(1+x) \times 10^4 x = 214 \times 10 \times 10$$

$$M = \frac{1}{2} \times \frac{214 \times 10^4 \times 0.12}{2} \times 1.2 = \pm 200 \times 10^4$$



However, this stress is computed for a cantilever bar with complete reversal of stress. In fact this section will not be so stressed because the shoulder of the pin will be set solidly against its seat, preventing this oscillating stresses from reaching this point of smallest cross section.



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## THEORETICAL CHECK OF SONNTAG REQUIREMENTS

It can be considered that the distortion is concentrated in the test specimen, and all other parts are of infinite rigidity. This is not true, but is sufficiently accurate to prove that the expected deflections are well within limitations of the Sonntag Universal Fatigue Testing Machine.

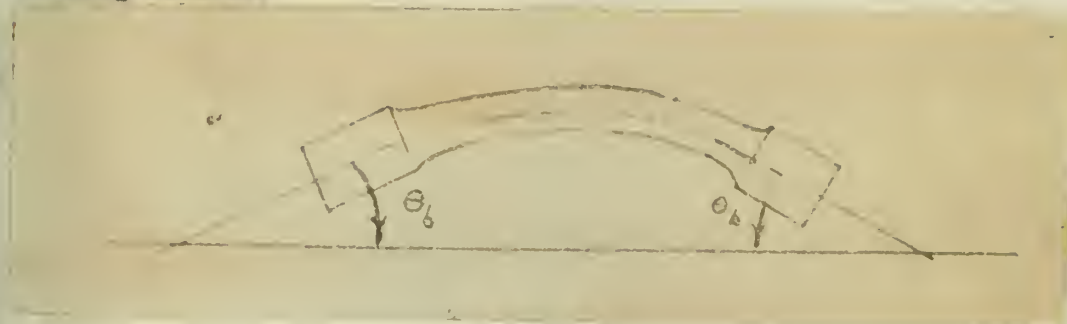


Figure 8

### Distortion of Specimen

The distortion of the test section can be divided into two parts. One part due to bending, and one part due to torsion. In each case the center section can be considered as fixed due to the symmetry of the jig.

For the distortion due to bending

$$\theta_b = \frac{1}{2} \frac{Ml}{EI_2} = \frac{1}{2} \frac{M \frac{13}{16} \times 64}{E \pi (\frac{1}{2})^4} + \frac{1}{2} \frac{M \frac{3}{8} \times 64}{E \pi (\frac{3}{4})^4} = 144.6 \frac{M}{E}$$

Similarly the angle of twist from the center section is

$$\theta_t = \frac{1}{2} \frac{Mt \ 32l}{G \pi d^4} = \frac{Mt}{G} \left[ \frac{32 \times \frac{13}{16}}{\pi (\frac{1}{2})^4} + \frac{32 \times \frac{3}{8}}{\pi (\frac{3}{4})^4} \right] = 144.6 \frac{Mt}{G}$$

The deflection experienced by the machine is relative vertical motion between the central self-aligning bearing and the hinge pins

# THEORETICAL BASIS OF SPINNING

It can be considered that the material is concentrated in the  
 test specimen, and all other parts are of infinite rigidity. This is  
 not true, but it sufficiently approximates to prove that the expected de-  
 flexions are well within limitations of the spinning machine design.  
 Testing Machine.

Figure 2

## Distribution of Specimen

The distribution of the test specimen can be divided into two parts.  
 One part due to bending, and one part due to torsion. In each case  
 the center section can be considered as fixed due to the quantity of  
 the life.

$$\theta = \frac{1}{2} \frac{M L}{E I} = \frac{1}{2} \frac{M L}{E \pi (\frac{D}{4})^4} + \frac{1}{2} \frac{M L}{E \pi (\frac{D}{4})^4} = 1440 \frac{M}{E}$$

$$\theta = \frac{1}{2} \frac{M L}{E \pi b^4} = \frac{M L}{2 E \pi b^4} \left[ \frac{2 \pi \times 10^6}{\pi (\frac{D}{4})^4} + \frac{2 \pi \times 10^6}{\pi (\frac{D}{4})^4} \right] = 1440 \frac{M L}{E}$$

The deflection experienced by the machine is relative vertical  
 motion between the central self-aligning bearing and the blade pins



at the extremity of the arms. For the case of pure bending of a steel specimen where it is desired to have an oscillating tensile stress of 100,000 PSI

$$M = \frac{6 I}{c} = \frac{100,000 \pi \left(\frac{1}{2}\right)^3}{32} = 1,230 \text{ lb. in.}$$

The length of the arm is 7.668 in.

$$\text{so force of platen } P_0 = 2X \frac{1,230}{7.668} = 321 \text{ lb.}$$

and the deflection of platen

$$\delta_s = \pm \left[ 144.6 \times \frac{1230}{30^6} \right] 7.668 = \pm 0.0455 \text{ in.}$$

Similarly if aluminum is subjected to the same loading

$$\delta_{al} = \pm 0.124 \text{ in.}$$

If it is desired to stress the specimen simultaneously with 100,000 PSI tensile stress at  $\pm$  50,000 PSI shear stress the following will be the case

$$\tau = \frac{16 M_x}{\pi d^3}$$

$$M_x = \frac{50,000 \pi \left(\frac{1}{2}\right)^3}{16} = 1,230 \text{ lb in.}$$

First it is necessary to find the arm angle and force P.

$$M = (1.668 \ 6 \cos \chi) P = 1230 \text{ lb. in.}$$

$$M_t = 6 (\sin \chi) P = 1230 \text{ lb. in.}$$

$$\chi = 56.5^\circ$$

$$P = 246 \text{ lb.}$$



at the extremity of the beam. For the case of pure bending of a beam specimen where it is desired to have an oscillating variable stress of

$$100,000 \text{ PSI}$$

$$\sigma = \frac{EI}{\rho} = \frac{100,000 \pi \left(\frac{1}{2}\right)^2}{32} = 1,570 \text{ in.}$$

The length of the arm is 7.643 in.

$$\text{so force of plates } P = \frac{1,570}{7.643} = 205 \text{ lb.}$$

and the deflection of plates

$$\delta = \pm \left[ 144.6 \times \frac{1230}{30^2} \right] 7.643 = \pm 0.0422 \text{ in.}$$

Statically it is assumed to be subjected to the same loading

$$\delta = \pm 0.124 \text{ in.}$$

If it is desired to stress the specimen simultaneously with 100,000 PSI tensile stress at 10,000 PSI shear stress the following will be the case

$$M = \frac{20,000 \pi \left(\frac{1}{2}\right)^2}{32} = 1,570 \text{ lb in.}$$

$$V = \frac{10,000 \pi \left(\frac{1}{2}\right)^2}{32} = 785 \text{ lb in.}$$

First it is necessary to find the arm angle and force P.

$$M = (1.668 \cos \alpha) P = 1,570 \text{ lb in.}$$

$$V = (1.668 \sin \alpha) P = 785 \text{ lb in.}$$

$$\alpha = 26.5^\circ$$

$$P = 216 \text{ lb.}$$

Therefore the machine must be set for  $\gamma = 56.5^\circ$

And force  $P_0 = 492 \text{ lb.}$

$$\therefore \delta_s = \pm \left\{ \left[ 144.6 \times \frac{1230}{3016} \times \frac{1230}{246} \right] + \left[ 144.6 \times \frac{1230}{1116} \times \frac{1230}{246} \right] \right\}$$

$$\delta_s = \pm 0.1104 \text{ in.}$$

similarly for aluminum

$$\delta_{al} = \pm 0.30 \text{ in.}$$

If it is desired to subject a steel specimen to a shear stress of 50,000 PSI, again  $P$  and  $\gamma$  must be determined

$$M = (1.668 + 6 \cos \gamma) P = 0 ; M_t = 6(\sin \gamma) P = 1230 \text{ lb. in.}$$

$$\gamma = 106.3^\circ \quad P = 214 \text{ lb.}$$

so setting of machine will be  $\gamma = 106.3^\circ$

and force  $P_0 = 428 \text{ lb.}$

All of these computations assume no internal damping in the machine or in the specimen. As stated before this will be investigated for accuracy at a later date.

To further justify neglecting the motion of the outer hinge first consider the inertia force imparted by the arm and link on the specimen. By considering the mass of the links concentrated at the ends of links as outlined by Den Hartog (2) in "Mechanical Vibrations" and finding the accelerations of the ends neglecting the residual inertia torque, the inertia forces can be computed. As the axial inertia force is greatest in the pure bending configuration, this will be investigated.

The concentrated masses will be:

Theory of the machine will be for  $\gamma = 104.3^\circ$

$$\left\{ \begin{aligned} & 144.6 \times \frac{12.50}{11.5} \times \frac{0.551}{1.5} \\ & + \left[ \frac{12.50}{3.45} \times \frac{0.551}{3.015} \times \frac{12.50}{11.5} \right] \end{aligned} \right\} \frac{12.50}{1.5} = 1$$

$$\gamma_2 = 104.3^\circ$$

Similarly for aluminum

$$\gamma_{al} = 104.3^\circ$$

If it is desired to subject a wheel specimen to a shear stress of

20,000 PSI, again  $P$  and  $\gamma$  must be determined

$$N = (1.66 + 0.005 \cos \gamma) P = 0; \quad \gamma = 6(\sin \gamma) P = 1250 \text{ lb. in.}$$

$$P = 104.3^\circ$$

so setting of machine will be  $\gamma = 104.3^\circ$

and force  $P = 1250 \text{ lb.}$

All of these computations assume no lateral loading in the machine

or in the specimen. As stated before this will be investigated for

specimen at a later date.

To further justify neglecting the motion of the overhanging flange

consider the inertia force imparted to the rim and flange on the specimen.

By considering the mass of the flange connected at the ends of links

as outlined by Dr. Langer (2) in "Mechanical Vibrations" and finding

the accelerations of the ends neglecting the residual inertia torque,

the inertia forces can be computed. As the wheel inertia force is

present in the pure bending condition, this will be investigated.

The concentrated masses will be:





Figure 9

### Schematic of Jig

for a vertical displacement of platen of  $0.3 \sin 60\pi t$  in., the horizontal displacement of the bearing will be  $l[1 - \cos \theta]$

where

$$\theta = \frac{0.3}{7.668} \sin 60\pi t = 0.039 \sin 60\pi t \text{ radians}$$

expanding into a series

$$l[1 - \cos \theta] = l \left[ \frac{\theta^2}{2!} - \frac{\theta^4}{4!} + \frac{\theta^6}{6!} - \frac{\theta^8}{8!} \dots \right]$$

$$= 7.668 \left[ \frac{0.0015 \sin^2 60\pi t}{2} - \frac{0.0000023 \sin^4 6\pi t}{24} \dots \right]$$

considering only the first term as significant the equations of the motion of the pivot are:

$$y = 0.0058 \sin^2 60\pi t = 0.0058 \sin^2 188t$$

$$y = 2.18 \sin 188t \cos 188t$$

$$y = 400 \cos^2 188t - 400 \sin^2 188t = 400 \cos 376t$$

The axial inertia force exerted upon the specimen will then be

$$\begin{aligned} F_i &= - \frac{2.76}{386} \times 400 \cos 376t \\ &= - 2.95 \cos 376t \text{ lb.} \end{aligned}$$



10. To the

$$\sin 0.02 \approx 0.02 \text{ rad} = 0.02 \times \frac{180}{\pi} = 1.146^\circ$$

$$\left[ \therefore \frac{1}{2} - \frac{1}{2} + \frac{1}{2} - \frac{1}{2} \right] = 0$$

$$\left[ \frac{\dots \text{270}^{\circ} \text{min } 280000.0}{43} = \frac{\dots \text{270}^{\circ} \text{min } 2100.0}{5} \right] 8.255$$

1958 25.00 1959 20.00 1960 15.00

5881 1.16 20.0 = 5882 1.16 20.0 = 7

1951-1952 年 11 月 15 日

1960-1961

100-443887-100

$$4.5 = \frac{5.40}{2.80} \times 100 \text{ g}$$

21 - 22.5 - 23.5

Thus the specimen has a uniform compressive stress of 15 PSI the extremities of the stress stroke. In general the amplitude oscillation will be much less and the resultant inertia forces also be much less. However even the 15 PSI is within the limit of error of the fatigue testing in the range of stresses necessary to produce this deflection ( $\approx 240,000$  PSI for aluminum).

The inertia forces of the jig which are reflected back into the inertia force compensator must be within the limit of tuning of the machine, as the machine is presently able to accommodate any jig with an equivalent weight of 15.4 pounds. To check the expected equivalent weight of this jig in the pure bending position, the masses of the various links are considered concentrated at the pivot points as before. All parts of the jig which are rigidly attached to the platen will oscillate with it and so will have an equivalent mass equal to their mass. The equivalent mass then for the pure bending condition is

Aluminum in center	3.75
2 bearings (CFI)	0.80
Specimen	0.50
Weight of arm and specimen holder concentrated at bearing	6.65
	<hr/>
	11.70 lbs.

As the machine is tuned for an equivalent weight of 15.4 pounds then 3.7 pounds of added weight is necessary in this condition to tune

With the specimen in a vertical position of 15 PSI the  
 experiments of the above series. In general the specimens  
 will be more than the required length for the test.  
 However even the 15 PSI is about the limit of what can be  
 tested in the range of stresses necessary in testing this material  
 (2,500,000 PSI for aluminum).

The inertia forces of the 15 PSI are released into the  
 inertia force component must be within the limit of inertia of the  
 machine, as the machine is generally able to accommodate any 15  
 with an equivalent weight of 15.4 pounds. In such the repeated opera-  
 tion of the 15 PSI in the test position, the mass of the  
 varying inertia and component concentrated at the pivot point as be-  
 fore. All parts of the 15 PSI which are rigidly connected to the pivot  
 will oscillate with it and will have an equivalent mass equal to  
 their mass. The equivalent mass for the test position condition is

3.75	Equivalent to center
0.50	3 bearings (CWI)
0.10	Specimen
	Weight of arm and specimen below concentrated
4.45	at bearing
<hr/>	
11.70 lbs.	

As the machine is rated for an equivalent weight of 15.4 pounds  
 then 11.70 pounds of total weight is necessary in this condition to run

the machine to the synchronous frequency. The equivalent weight of this system will vary slightly from pure bending to pure torsion but will be well within the limits of compensation of the machine.



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the action will only slightly have been taken to  
will be well within the limits of cooperation of the nation.

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## ACTUAL CHECK OF OPERATING JIG

Upon completion of the jig it was assembled and operated with a specimen of negligible weight and rigidity to determine the equivalent weight of the jig in various configurations. This specimen consisted of a  $1/8"$  diameter steel rod with a brass adapter on each end to attach to the specimen holding bolts. Having the curve of equivalent weights, a steel specimen with SR-4 type A-7 strain gages, was installed and tested. These strain gages were located such that two were located along the axes of maximum bending stress, and two along the axes of maximum strain from torsion and along the zero bending strain axes. With this configuration the two bending gages were not effected by torsion and the two sheer gages were not effected by bending. These gages were connected to two channels of a Hathway oscillograph for dynamic strain recording. The other half of the bridge circuits consisted of Baldwin SR-4 Strain Calibration Units which facilitated the amplitude calibration for each run.

The machine was then operated in various ratios of bending and torsion and the results computed (see Appendix). The errors found can be attributed to the technique of testing. A check of the stresses found and the predicted stress would indicate that the torque arms were not positioned exactly. For example, consider the readings for  $\chi = 60^\circ$ . If it is assumed that the actual arm angle was  $59^\circ$ , the predicted bending stress and sheer stress now being  $\sigma_b = 19,350$  P.S.I. and  $\tau = 10,400$  P.S.I., then the error involved would be  $+1.8\%$  for bending and  $+0.95\%$  for torsion. This will indicate that the arms were not set exactly

# ANALYSIS OF CRACKS IN

Upon completion of the test the specimen was removed and examined with a specimen of variable weight and length to determine the relationship of weight of the lig in various configurations. This specimen consisted of a 1/8" diameter steel rod with a square shoulder on each end to attach to the specimen holding bolts. Having the same of equal weight, a steel specimen with 32-A type A-V strain gages, was installed and

tested. These strain gages were located such that two were located along the axis of maximum bending stress, and two along the axis of maximum strain from torsion and along the axis bending strain axis. When this configuration the two bending gages were not affected by torsion and the two shear gages were not affected by bending. These gages were connected to two channels of a battery recorder for dynamic strain recording. The other half of the bridge circuit consisted of identical 32-A Strain Galvanometer Units which furnished the amplification calibration for each run.

The machine was then operated at various rates of bending and torsion and the results compared (see Appendix). The stress found can be attributed to the technique of testing. A check of the stress found and the predicted stress would indicate that the technique was not positioned correctly. For example, consider the frequency for  $\lambda = 0.5$ . If it is assumed that the actual stress was 100, the predicted bending stress and shear stress was being  $\sigma = 10,000$  P.S.I. and  $\tau = 10,000$  P.S.I., then the stress involved would be 1.41 for bending and 1.41 for torsion. This will indicate that the stress was not correctly



throughout the test and the actual error involved is much less than indicated. Further testing and more accurate positioning of these arms will undoubtedly reduce these errors considerably.



Throughout the test and the several errors involved it was found  
indicated. Further testing will show errors' position of these  
and will undoubtedly reveal these errors completely.

It is to be noted that the errors in the test were not  
of the same nature as those in the test of the first group.  
The errors in the test of the first group were of the nature  
of errors of omission, while the errors in the test of the  
second group were of the nature of errors of commission.  
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## BIBLIOGRAPHY

1. A.S.T.M. Manual of Fatigue Testing.
2. Den Hartog. Mechanical Vibrations  
McGraw-Hill. 1947.
3. Dolan, T. J. Influence of Shape and Cross Section on  
Flexural Fatigue Strength of Metals.  
A.S.M.E. Transactions 72, July 1950.
4. Fuller, E. B. & Oberg, T. T. Fatigue Characteristics  
of Rotating-Beam versus Rectangular Cantilever Specimens  
of Steel and Aluminum Alloys.  
Proceedings A.S.T.M. Vol. 47, Page 665 (1947).
5. Gensamer, M. Strength of Materials under Combined Stresses.
6. Gough, H. J., Pollard, H. V. & Clenshaw, W. J. Some  
Experiments on the Resistance of Metals to Fatigue  
under Combined Stresses.  
Aeronautical Research Council. R&M 2522.
7. Hamm, C. W., & Crane, E. J. Mechanics of Machinery.  
McGraw-Hill. 1938.
8. Lazan, B. J. Fatigue-Testing Machine.  
Machine Design. May 1947.
9. Lazan, B. J. Effect of Damping Constants and Stress  
Distribution on the Resonance Response of Members.  
A.S.M.E. Paper 52-A-8.
10. Marin, J. Engineering Materials.  
Prentice Hall. 1952.
11. Moore, H. F. The Effect of Type of Testing Machine on  
Fatigue Test Results.  
Proceedings A.S.T.M. 1941. Page 133.
12. Patterson, F. G. Factors when Considering Anti-Friction  
Bearings.  
Product Engineering. Oct. 1950. Pages 141-143.
13. Roos, P. K., Lemmon, D. C., & Ranson, J. T. Influence  
of Type of Machine, Range of Speed, and Specimen Shape  
on Fatigue Test Data.  
A.S.T.M. Bulletin. May 1949, Pages 63-65.

# REFERENCES

1. A.S.T.M. Manual of Fatigue Testing.
2. Ben Harzog. Mechanical Vibrations McGraw-Hill. 1947.
3. Bolan, T. J. Influence of Stress and Cross Section on Flexural Fatigue Strength of Metals. A.S.T.M. Transactions Vol. 100, 1940.
4. Bolan, T. J. & O'Neil, E. T. Fatigue Characteristics of Rotating Bars versus Geometrical Concentration of Stress and Material Allotropy. Proceedings A.S.T.M. Vol. 47, Page 665 (1947).
5. Gensamer, M. Strength of Materials under Combined Stresses.
6. Smith, N. J., Polak, H. F. & Glasstone, K. J. Some Experiments on the Fatigue of Metals in Fatigue under Combined Stresses. Aeronautical Research Council. R.M. 2524.
7. Hume, C. W. & Crane, E. J. Mechanics of Machinery. McGraw-Hill. 1930.
8. Lamm, B. J. Fatigue-Testing Machinery. Machine Design. May 1947.
9. Lamm, B. J. Effect of Design Constants and Stress Distribution on the Rotational Lives of Shafts. A.S.T.M. Paper 22-A-51.
10. Martin, J. Engineering Materials. Prentice Hall. 1932.
11. Moore, N. F. The Effect of Type of Testing Machine on Fatigue Test Results. Proceedings A.S.T.M. 1941. Page 131.
12. Patterson, V. O. Factors when Considering Anti-Fatigue Design. Product Engineering. Oct. 1930. Pages 141-143.
13. Root, P. K., Lamm, B. J., & Hanson, V. E. Influence of Type of Machine, Range of Stress, and Specimen Shape on Fatigue Test Data. A.S.T.M. Bulletin. May 1948. Pages 42-52.



14. Sauer, J. A. A Study of Fatigue Phenomenon under Combined Stress .  
Proceedings at the 7th International Contress for Applied Mechanics. 1948.
15. Yorgiadis, A. Non-Linear Specimens in Inertia-Force Compensator Type of Fatigue Machine.  
Baldwin Locomotive Works. Report 90823-S.



1A. Smay, J. A. A Study of Foreign Investment in the United States.  
Stress.  
Proceedings of the 10th International Congress for Applied  
Economics, 1946.

1B. Yonjastik, A. Non-linear Spectra in Linear-Force Systems.  
The Journal of Applied Mechanics.  
Salada International House, Report 1946-2.

## APPENDIX

### TRANSIENTS

One of the greatest drawbacks to the inertia force type of Fatigue Testing machines is the presence of transients while bringing the testing machine up to synchronous speed. In general by the time that the transients have died out the test specimen has an unknown history or work hardening.

Perhaps, one method of eliminating or reducing this phenomenon would be the addition of a magnetic damping circuit. This would consist of a strong magnetic field through the platen while the motor is being brought up to speed. The magnetic field could then be broken at a time that the force is passing through neutral, or reduced gradually depending upon the best procedure as determined by test. A simple electro-magnet could easily be installed inside the cabinet to do just this function.

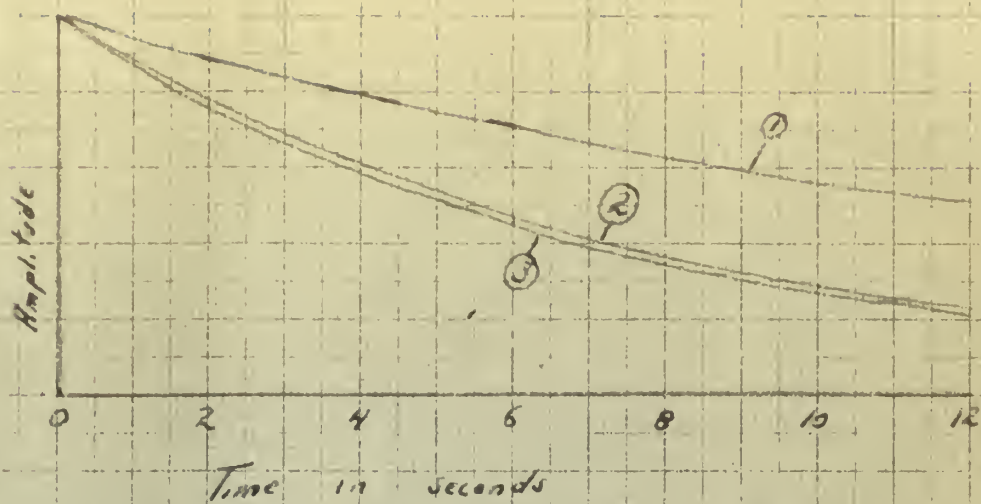
# APPENDIX

## TRANSISTORS

One of the greatest drawbacks to the transistor type of amplifier is the presence of parasitic oscillations which limit the useful frequency up to approximately 100 Mc. In general it is the fact that the transistors have not yet been developed to an extent which is satisfactory for work in this field.

Perhaps, one method of eliminating or reducing this phenomenon would be the addition of a magnetic field to the circuit. This would assist in a strong magnetic field through the plates while the motor is being brought up to speed. The magnetic field could then be broken at a time that the force is needed through neutral, or reduced gradually depending upon the best procedure as determined by test. A simple electro-magnet could easily be installed inside the cabinet in its just this function.





Damping Envelope of Sonntag Machine

- ① Machine alone  $\kappa = 0.037$
- ② Machine with standard Torsion  $\kappa = 0.125$   
Fig without specimen
- ③ Machine with standard Torsion  $\kappa = 0.130$   
Fig with Aluminium Specimen

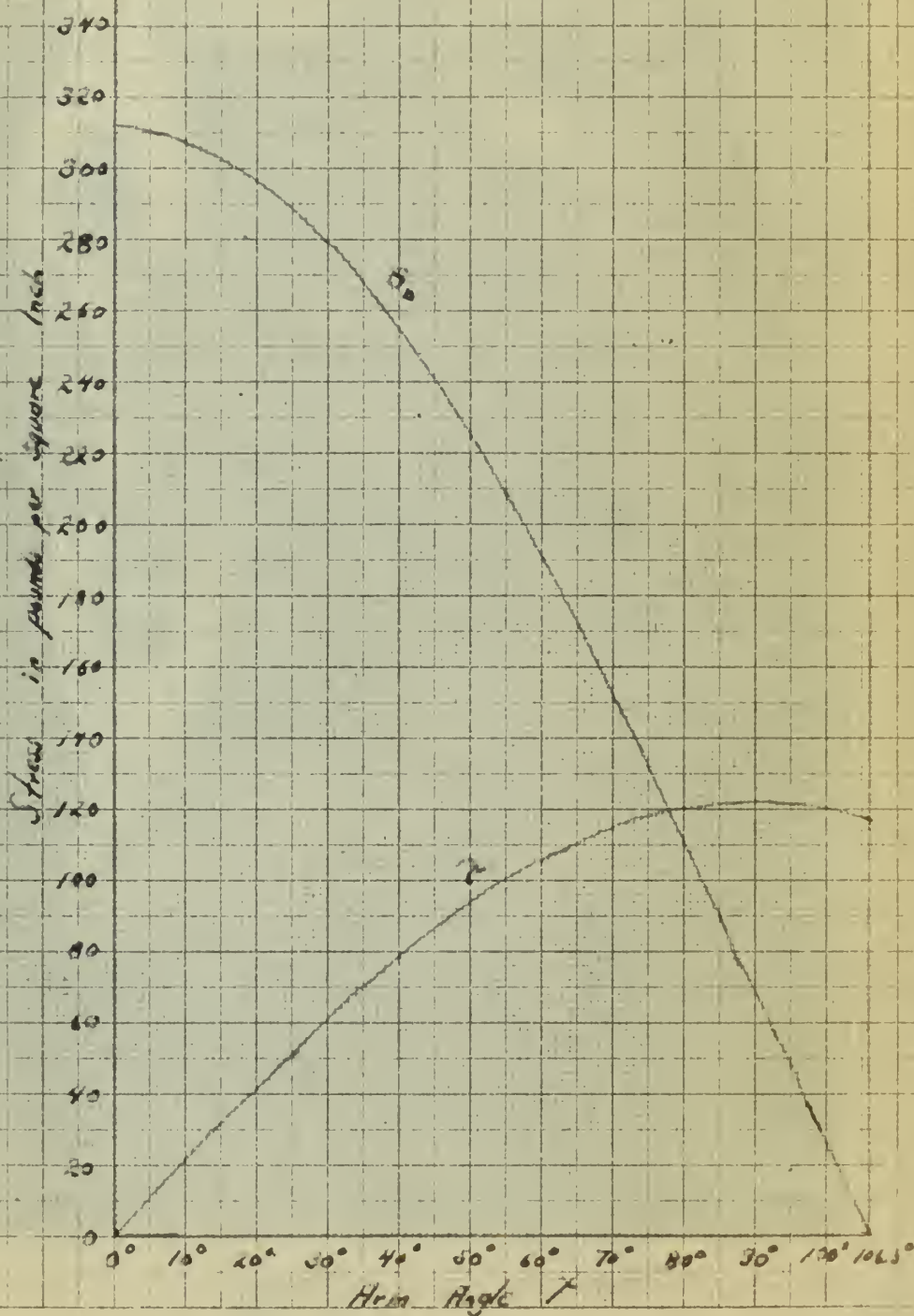
Where  $A = A_0 e^{-\kappa t}$





Maximen Tensile Stress due to Bending,  $\sigma_b$   
and Shear Stress due to Torsion,  $\tau$   
Vs.

Hrm angle  $\theta$   
per Posed of static or Dynamic Force  
For a 0.500 inch dia specimen



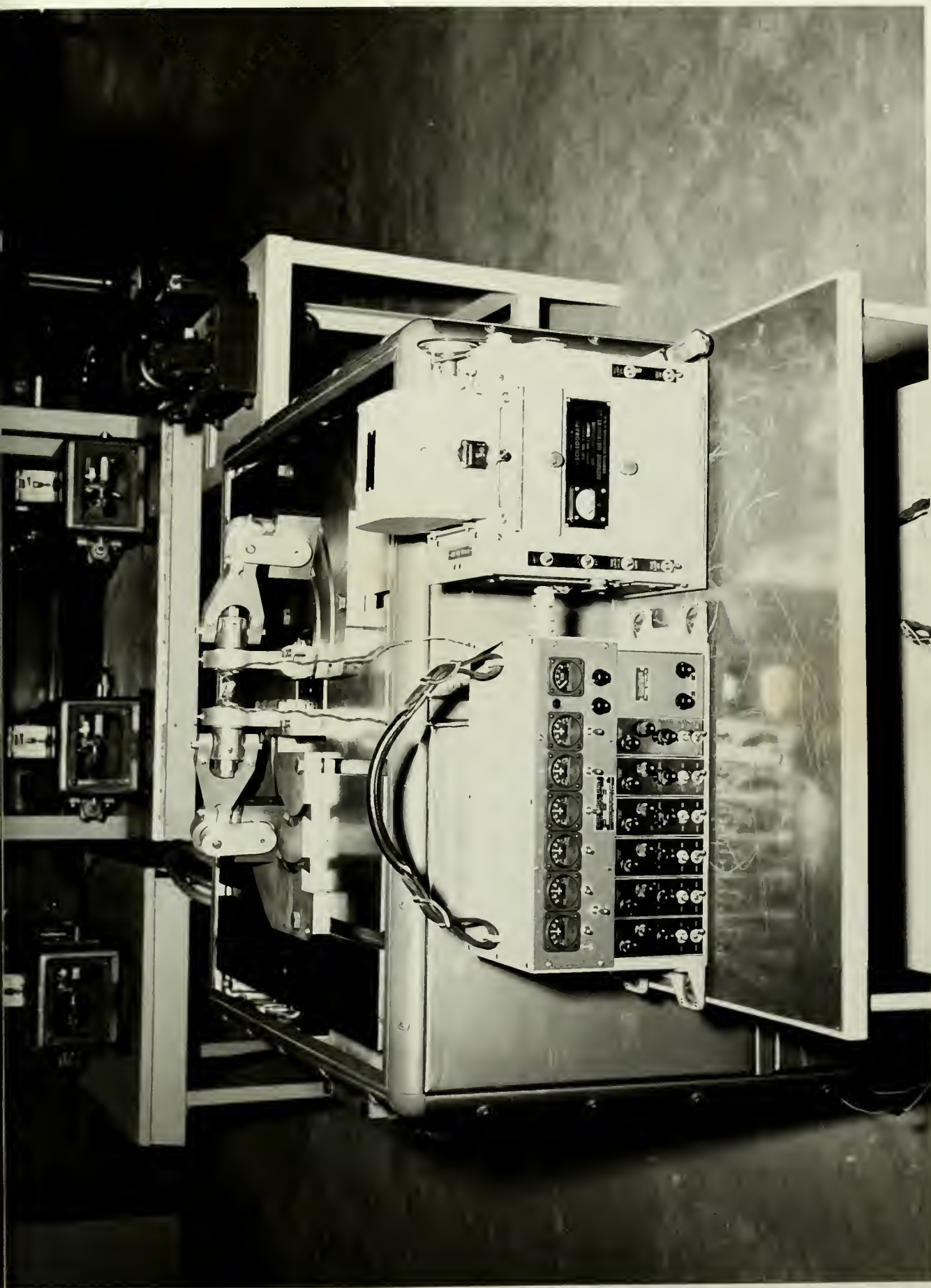




ASSEMBLED JIG



DATE \_\_\_\_\_  
I hereby authorize the \_\_\_\_\_  
to make \_\_\_\_\_  
of the \_\_\_\_\_  
not to be used for publication



ASSEMBLED JIG SHOWING INSTRUMENTATION





$\chi$	$l_{60}$	$l_{70}$	$M$	$M_t$	$\sigma$	$\tau$
$0^\circ$	7.6855	0	3.8427	0	312.0	0
$10^\circ$	7.5355	1.042	3.7977	0.5210	308.0	21.2
$20^\circ$	7.3255	2.050	3.6627	1.0250	297.5	41.8
$30^\circ$	6.8805	3.000	3.4403	1.5000	279.5	61.0
$40^\circ$	6.2805	3.855	3.1403	1.9275	255.0	78.5
$50^\circ$	5.5405	4.955	2.7703	2.4775	225.0	93.5
$60^\circ$	4.6855	5.195	2.3427	2.5975	190.1	105.6
$70^\circ$	3.7355	5.640	1.8677	2.8200	151.7	114.8
$80^\circ$	2.7275	5.910	1.3637	2.9500	111.0	120.2
$90^\circ$	1.6855	6.000	0.8427	3.000	68.4	122.0
$100^\circ$	0.6435	5.910	0.3217	2.9500	26.2	120.2
$106.3^\circ$	0	5.755	0	2.8775	0	117.0



5	0	1/4	1/2	3/4	1	1 1/4
0	0.812	0	0.812	0	1.625	0
2.5	0.812	0.812	0.812	0.812	1.625	0.812
5.0	0.812	0.812	0.812	0.812	1.625	0.812
7.5	0.812	0.812	0.812	0.812	1.625	0.812
10.0	0.812	0.812	0.812	0.812	1.625	0.812
12.5	0.812	0.812	0.812	0.812	1.625	0.812
15.0	0.812	0.812	0.812	0.812	1.625	0.812
17.5	0.812	0.812	0.812	0.812	1.625	0.812
20.0	0.812	0.812	0.812	0.812	1.625	0.812
22.5	0.812	0.812	0.812	0.812	1.625	0.812
25.0	0.812	0.812	0.812	0.812	1.625	0.812
27.5	0.812	0.812	0.812	0.812	1.625	0.812
30.0	0.812	0.812	0.812	0.812	1.625	0.812
32.5	0.812	0.812	0.812	0.812	1.625	0.812
35.0	0.812	0.812	0.812	0.812	1.625	0.812
37.5	0.812	0.812	0.812	0.812	1.625	0.812
40.0	0.812	0.812	0.812	0.812	1.625	0.812
42.5	0.812	0.812	0.812	0.812	1.625	0.812
45.0	0.812	0.812	0.812	0.812	1.625	0.812
47.5	0.812	0.812	0.812	0.812	1.625	0.812
50.0	0.812	0.812	0.812	0.812	1.625	0.812
52.5	0.812	0.812	0.812	0.812	1.625	0.812
55.0	0.812	0.812	0.812	0.812	1.625	0.812
57.5	0.812	0.812	0.812	0.812	1.625	0.812
60.0	0.812	0.812	0.812	0.812	1.625	0.812
62.5	0.812	0.812	0.812	0.812	1.625	0.812
65.0	0.812	0.812	0.812	0.812	1.625	0.812
67.5	0.812	0.812	0.812	0.812	1.625	0.812
70.0	0.812	0.812	0.812	0.812	1.625	0.812
72.5	0.812	0.812	0.812	0.812	1.625	0.812
75.0	0.812	0.812	0.812	0.812	1.625	0.812
77.5	0.812	0.812	0.812	0.812	1.625	0.812
80.0	0.812	0.812	0.812	0.812	1.625	0.812
82.5	0.812	0.812	0.812	0.812	1.625	0.812
85.0	0.812	0.812	0.812	0.812	1.625	0.812
87.5	0.812	0.812	0.812	0.812	1.625	0.812
90.0	0.812	0.812	0.812	0.812	1.625	0.812
92.5	0.812	0.812	0.812	0.812	1.625	0.812
95.0	0.812	0.812	0.812	0.812	1.625	0.812
97.5	0.812	0.812	0.812	0.812	1.625	0.812
100.0	0.812	0.812	0.812	0.812	1.625	0.812

# EQUIVALENT WEIGHT

$\chi$	Comp. Wt.	Equ. Wt.
0°	4.4	11.0
15°	4.6	10.6
30°	5.2	10.2
45°	5.6	9.8
60°	5.8	9.6
90°	6.0	9.4
106.3°	6.1	9.3

THE UNIVERSITY OF CHICAGO

Time	Latitude	Longitude	Altitude	Temperature	Humidity	Wind Speed	Wind Direction	Clouds	Visibility	Remarks
0000	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0100	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0200	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0300	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0400	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0500	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0600	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0700	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0800	34.00	118.00	100	50.0	80	10	090	0	10	Clear
0900	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1000	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1100	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1200	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1300	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1400	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1500	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1600	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1700	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1800	34.00	118.00	100	50.0	80	10	090	0	10	Clear
1900	34.00	118.00	100	50.0	80	10	090	0	10	Clear
2000	34.00	118.00	100	50.0	80	10	090	0	10	Clear
2100	34.00	118.00	100	50.0	80	10	090	0	10	Clear
2200	34.00	118.00	100	50.0	80	10	090	0	10	Clear
2300	34.00	118.00	100	50.0	80	10	090	0	10	Clear



# STRESSES MEASURED ON SPECIMEN

$\chi$	Calib. Bending	Calib. Torsion	Amp Bending	Amp Torsion	Calib. Bending	Calib. Torsion	Amp Bending	Amp Torsion
0°			2.71		1.034		1.06	
30°(1)	0.800	0.945	3.075	0.98	0.96	0.259	0.985	0.265
30°(2)	0.76	1.00	2.88	1.02	0.948	0.255	0.973	0.262
60°	1.10	1.30	2.86	2.34	0.650	0.451	0.667	0.463
90°	1.06	1.36	0.95	2.80	0.224	0.515	0.230	0.529
106.3°	1.05	1.40	0.15	2.79	0.0357	0.497	0.0366	0.510

Assuming  $E = 29.5 \times 10^6 \text{ lb/in}^2$

$\mu = 0.3$

	Predicted				% error	
$\chi$	$\sigma_x$	$\tau_{xy}$	$\sigma_y$	$\tau_{xy}$	$\sigma_x$	$\tau_{xy}$
0°	31,300	0	31,200	0	+ 0.32	-
30°(1)	29,100	6,010	27,950	6,100	+ 4.1	- 1.5
30°(2)	28,700	5,950	27,950	6,100	+ 2.6	- 0.25
60°	19,700	10,500	19,010	10,560	+ 3.5	- 0.57
90°	6,800	12,000	6,840	12,200	- 0.58	- 1.6
106.3°	1,080	11,600	0	11,700	(+)	- 0.94

Data obtained using

Baldwin SR-4 Strain Gages Type AR-7

Hathaway Oscillograph Type 3-15-B

Baldwin SR-4 Calibration Units



# STRAIN GAGES ON CONCRETE

Gage 1				Gage 2				Gage 3				Gage 4			
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0

Assuming  $\epsilon = 10^{-4}$   $\epsilon = 10^{-4}$

Strain

Gage 1				Gage 2				Gage 3				Gage 4			
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0
10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0	10.0	1.0	0.0	0.0

Data obtained using

Model 80-4 Strain Gage Type A-1

Model 80-4 Strain Gage Type A-1

Model 80-4 Strain Gage Type A-1

# PARTS LIST

Part No.	Name	No. Req'd.	Material	Specification
1	Bearing Holder (fixed)	1	24ST	See Dwg.
2	Bearing Holder (oscillating)	1	24ST	See Dwg.
3	Bearing Support	1	24ST	See Dwg.
4	Bearing Support	1	24ST	See Dwg.
5	Specimen Holder	2	H.T. Steel	See Dwg.
6	Specimen Holder Sleeve	2	H.T. Steel	See Dwg.
7	Specimen Holder Cap	2	H.T. Steel	See Dwg.
8	Torque Pin	4	Drill Rod	See Dwg.
9	Bearing	2		Torrington 20NBK2040YZP
10	1/2 Bolt	2	H.T. Steel	1/2"X20X2" Hex.
11	5/16" Key	4	H.T. Steel	
12	Grease Fitting	8		1/8" Press Fit
13	Torque Arm	2	61ST	See Dwg.
14	Bearing	4		Torrington 12NBKL830YZP
15	Hold Down Pin	4	H.T. Steel	See Dwg.
16	Hold Down Link	4	61ST	See Dwg.
17	Lower Bearing Holder	2	H.T. Steel	See Dwg.
18	Lower Bearing Cap	2	H.T. Steel	See Dwg.
19	Table Clip	4	H.T. Steel	See Dwg.
20	Table Top	2	H.T. Steel	See Dwg.

# TABLE LIST

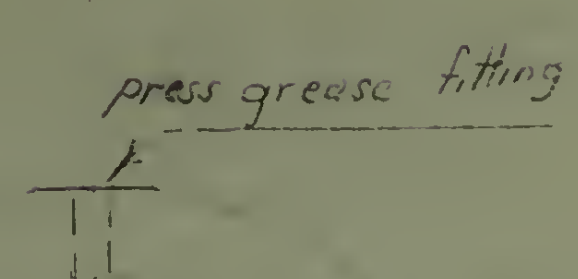
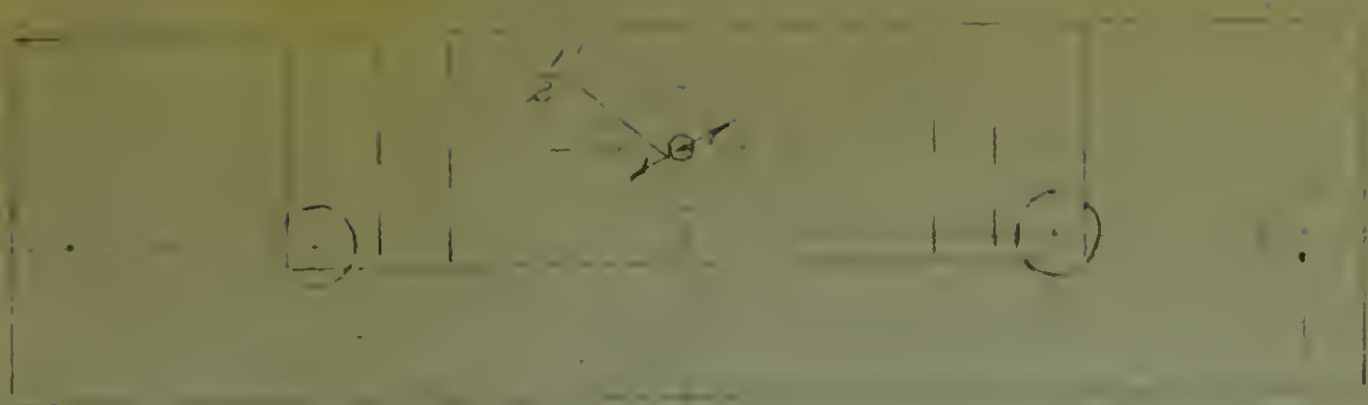
Part No.	Name	Qty.	Material	Special Location
1	Feeding Roller (Main)	1	Steel	See Dwg.
2	Feeding Roller (Auxiliary)	1	Steel	See Dwg.
3	Feeding Support	1	Steel	See Dwg.
4	Feeding Support	1	Steel	See Dwg.
5	Support Roller	2	H.T. Steel	See Dwg.
6	Support Roller Guide	2	H.T. Steel	See Dwg.
7	Support Roller Gap	2	H.T. Steel	See Dwg.
8	Support Pin	4	Drill Steel	See Dwg.
9	Feeding	1	Tool Steel	See Dwg.
10	1/2 Bolt	2	H.T. Steel	See Dwg.
11	3/16" Nut	4	H.T. Steel	See Dwg.
12	Grass Fitting	2	1/2" Iron Pipe	See Dwg.
13	Support Arm	2	Steel	See Dwg.
14	Feeding	1	Tool Steel	See Dwg.
15	Roll Down Pin	2	H.T. Steel	See Dwg.
16	Roll Down Pin	2	Steel	See Dwg.
17	Lower Feeding Roller	2	H.T. Steel	See Dwg.
18	Lower Feeding Gap	2	H.T. Steel	See Dwg.
19	Table Clip	4	H.T. Steel	See Dwg.
20	Table Top	2	H.T. Steel	See Dwg.



21	5/16" Bolt	8	H.T. Steel	5/16"X24X2½" Hex
22	7/16" Locknut	2	H.T. Steel	7/16" NF Elastic Stopnut
23	1/4" Machine Screw	8	H.T. Steel	1/4"X20X1/2" Hex.
24	1/4" Bolt	4	H.T. Steel	1/4"X28X1½" Hex.
25	1/4" Bolt	2	H.T. Steel	1/4"X28X2" Hex.
26	1/4" Locknut	6	H.T. Steel	1/4" NF Elastic Stopnut
27	3/16" Machine Screw	3	H.T. Steel	3/16"X30X2½" Hex.
28	3/16" Locknut	3	H.T. Steel	3/16" NF Elastic Stopnut
29	3/8" Bolt	4	H.T. Steel	3/8"X24X2½" Socket Hex.
30	5/8" Bolt	6	H.T. Steel	5/8"X18X1½" Hex.
31	3/4" Bolt	1	H.T. Steel	3/4"X16X1½" Hex.



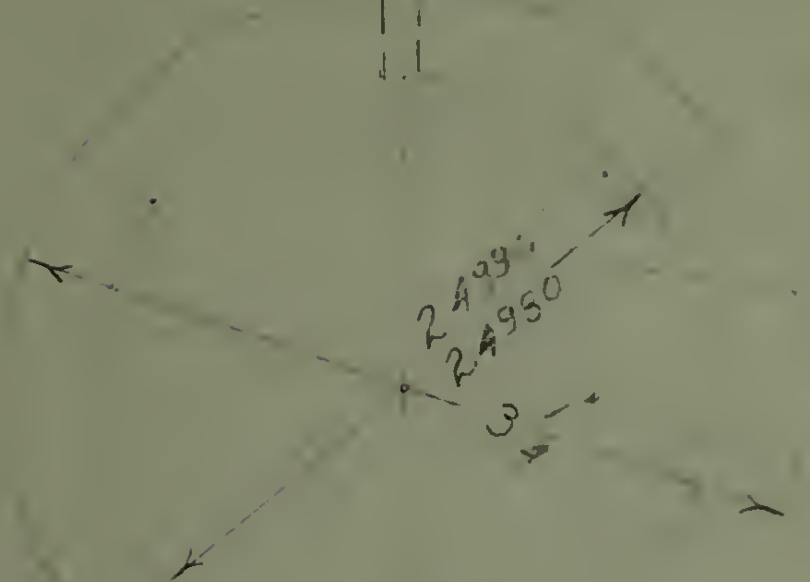
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23	2/10/1901	10	H.T. Street	2/10/1901	10
24	2/10/1901	10	H.T. Street	2/10/1901	10
25	2/10/1901	10	H.T. Street	2/10/1901	10
26	2/10/1901	10	H.T. Street	2/10/1901	10
27	2/10/1901	10	H.T. Street	2/10/1901	10
28	2/10/1901	10	H.T. Street	2/10/1901	10
29	2/10/1901	10	H.T. Street	2/10/1901	10
30	2/10/1901	10	H.T. Street	2/10/1901	10
31	2/10/1901	10	H.T. Street	2/10/1901	10
32	2/10/1901	10	H.T. Street	2/10/1901	10
33	2/10/1901	10	H.T. Street	2/10/1901	10
34	2/10/1901	10	H.T. Street	2/10/1901	10
35	2/10/1901	10	H.T. Street	2/10/1901	10
36	2/10/1901	10	H.T. Street	2/10/1901	10
37	2/10/1901	10	H.T. Street	2/10/1901	10
38	2/10/1901	10	H.T. Street	2/10/1901	10
39	2/10/1901	10	H.T. Street	2/10/1901	10
40	2/10/1901	10	H.T. Street	2/10/1901	10
41	2/10/1901	10	H.T. Street	2/10/1901	10
42	2/10/1901	10	H.T. Street	2/10/1901	10
43	2/10/1901	10	H.T. Street	2/10/1901	10
44	2/10/1901	10	H.T. Street	2/10/1901	10
45	2/10/1901	10	H.T. Street	2/10/1901	10
46	2/10/1901	10	H.T. Street	2/10/1901	10
47	2/10/1901	10	H.T. Street	2/10/1901	10
48	2/10/1901	10	H.T. Street	2/10/1901	10
49	2/10/1901	10	H.T. Street	2/10/1901	10
50	2/10/1901	10	H.T. Street	2/10/1901	10



1

It is a 'light' fit with Torrington #120 Sand

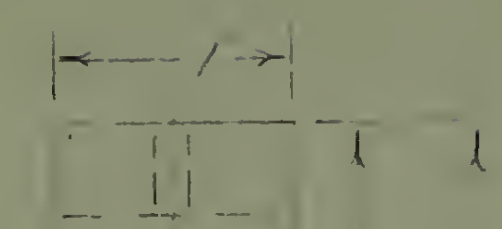
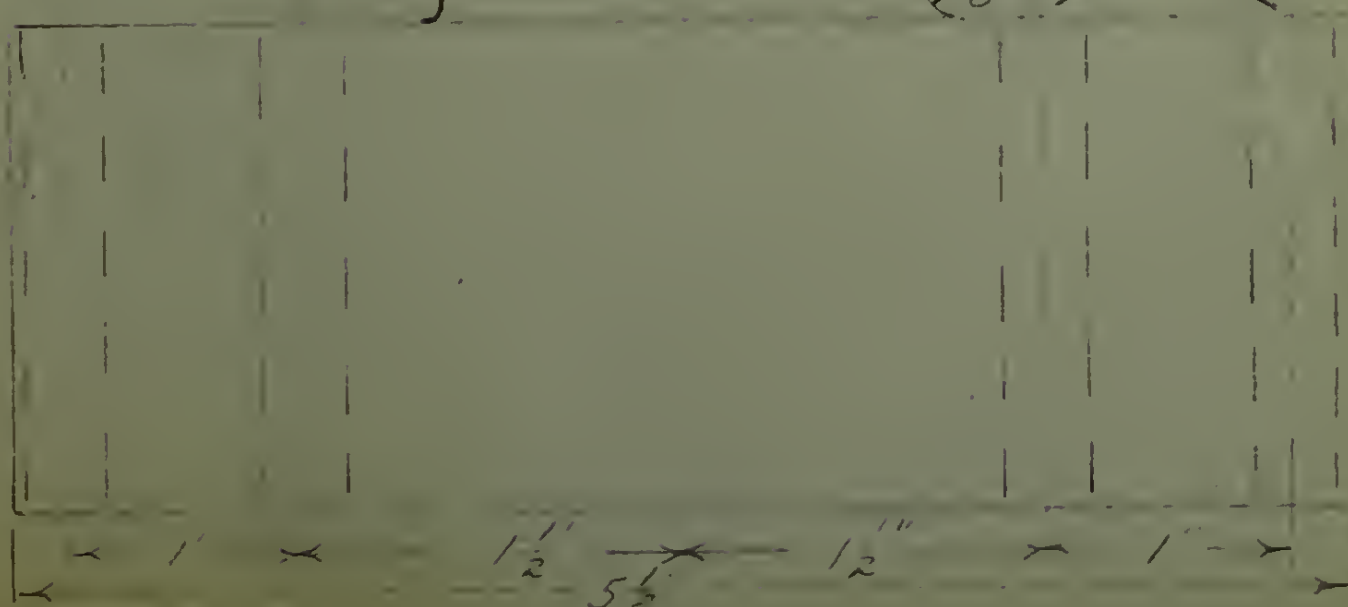
2A99  
2A950  
3



1/8" R

drill for 5/16" bolts

1/8" R



1/8"

X

3

5/16"

1/8" R



2"

# Notes

Material ~ 24ST-4

All tolerances  $\pm 0.0020$  except as specified

Fine machine finish throughout

One (1) required

U.S. Naval Postgraduate School  
Monterey, California

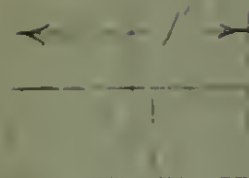
Combined Stress Fatigue T19

Center bearing (fixed)

AJ Lilmore

26 February 1957



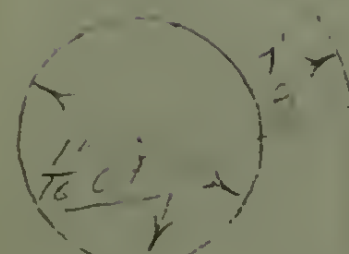


*Faint handwritten notes at the bottom of the page.*

2. 4981

3.

10



10000  
c 210  
Lancaster, Calif.

$$\frac{3}{2}$$

$\frac{7}{8} \times \frac{9}{10}$

1	7	7	7
0	0	0	0
1	1	1	1

Mer. cl ~ 245T-4

Fit tolerances  $\pm 0.020$  except as specified.

Fire marine fire throat

On (1) required

Central Street, New York, N. Y.

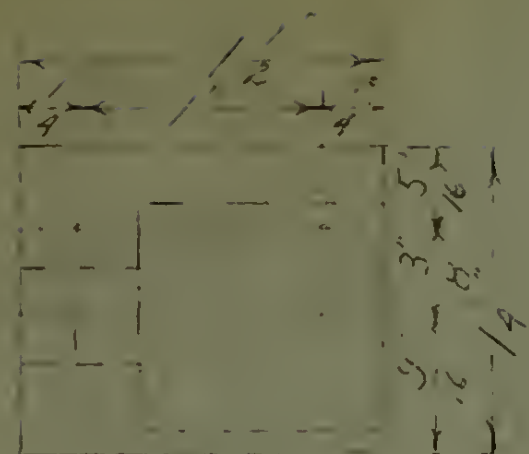
earing

MS Gilman 10

26 Dec 67 /

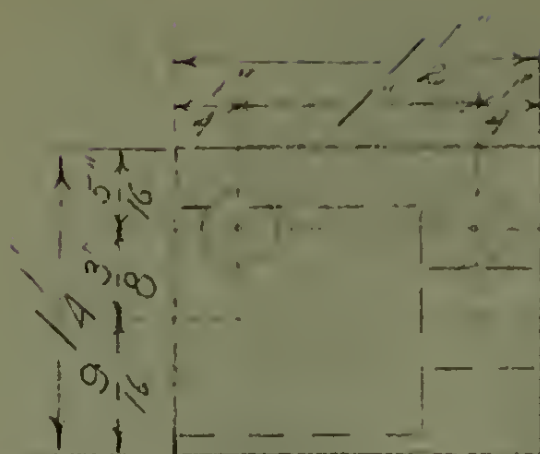




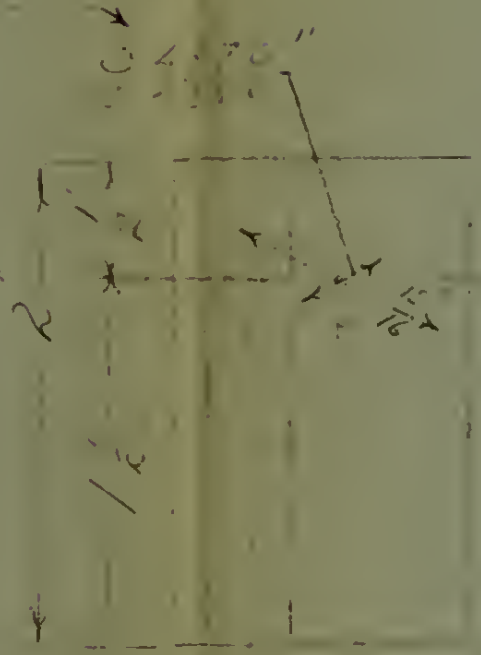
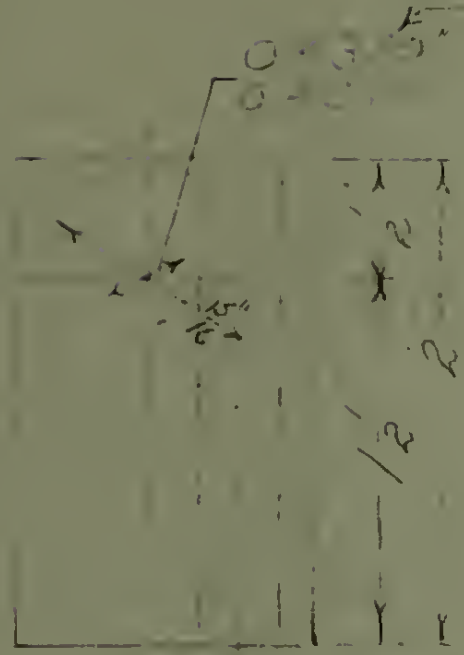
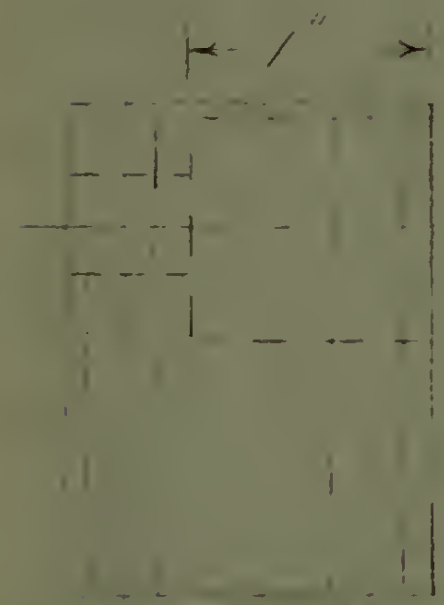


3

4



to 0.10" ft  
with  
M.P. CF-1



draw to  
scale

# Notes

Material ~ 24ST-4

All tolerances  $\pm 0.0020$  except as specified

Fine machine finish throughout

One (1) each required

U.S. Naval Postgraduate School  
Monterey, California

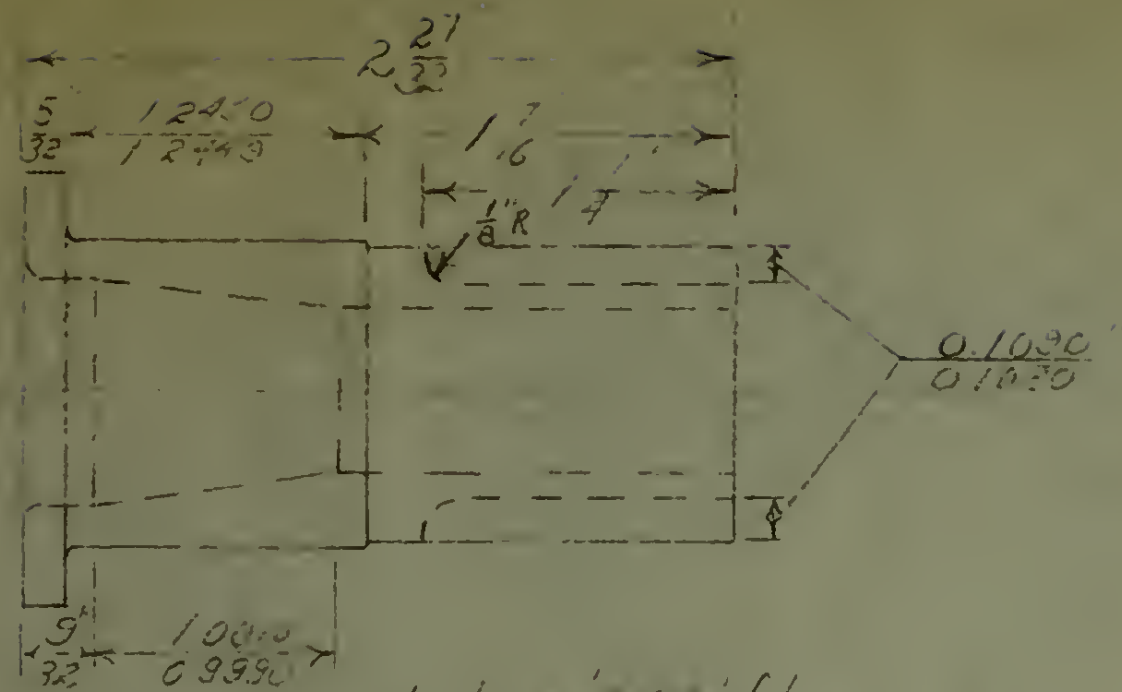
Combined Stress Fatigue Jig

Center bearing supports

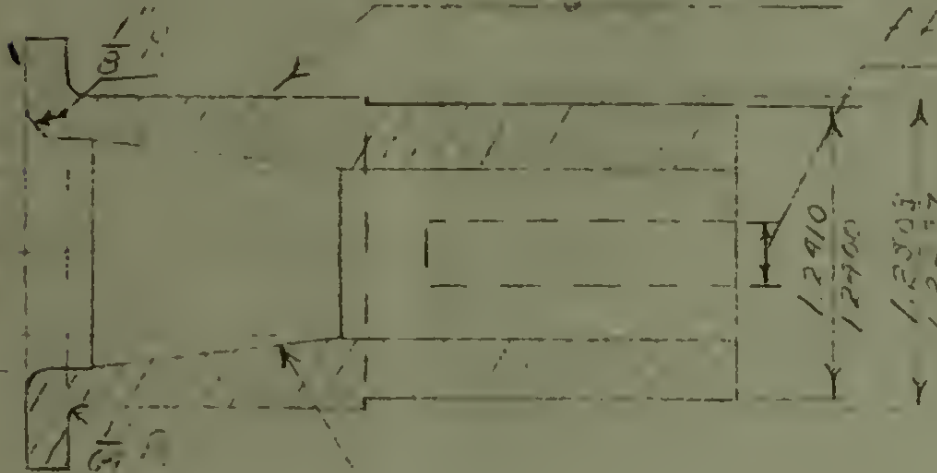
HJ Gilmore

27 February 1963

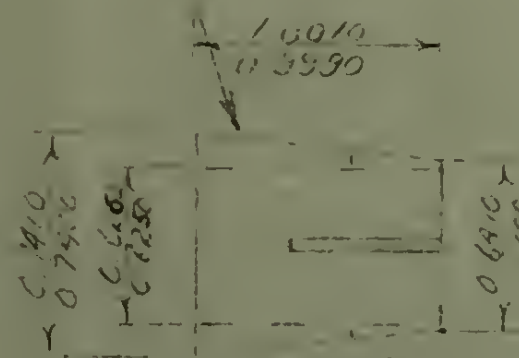




to be a 'press' fit  
with Torrington RT2050W to be a 'tight'  
fit with 5/16" hole



Typical machine  
Finish machine surfaces



5/16" wide slot  
from alternate end

Notes

Material ~ Cold-Rolled Steel

Fit tolerances  $\pm 0.002$  except as specified

Five machine finish surfaces, the other four are as-rolled

Two (2) each required

U.S. Naval Postgraduate School  
Monterey, California

Composite Stress Fatigue Test

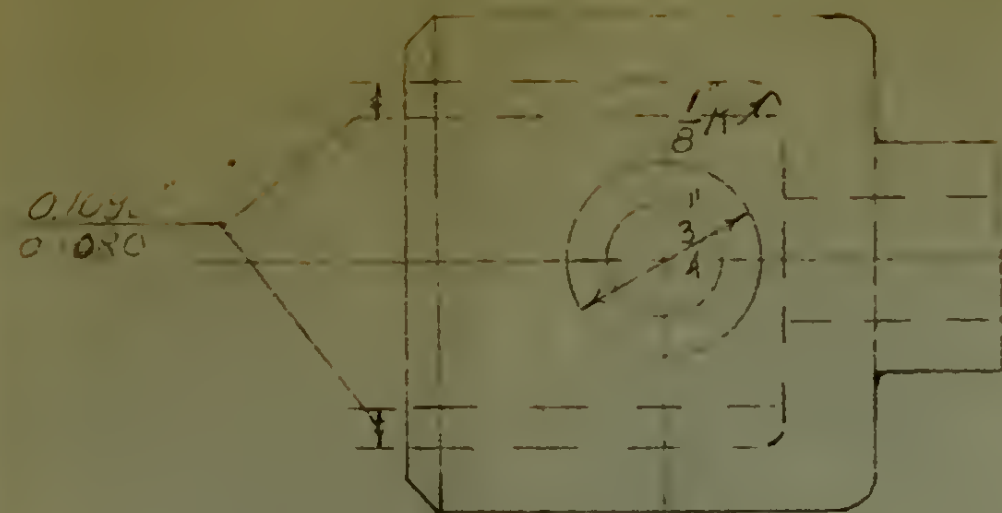
Specimen holder and clevis

H. J. L. Moore

27 February 1954

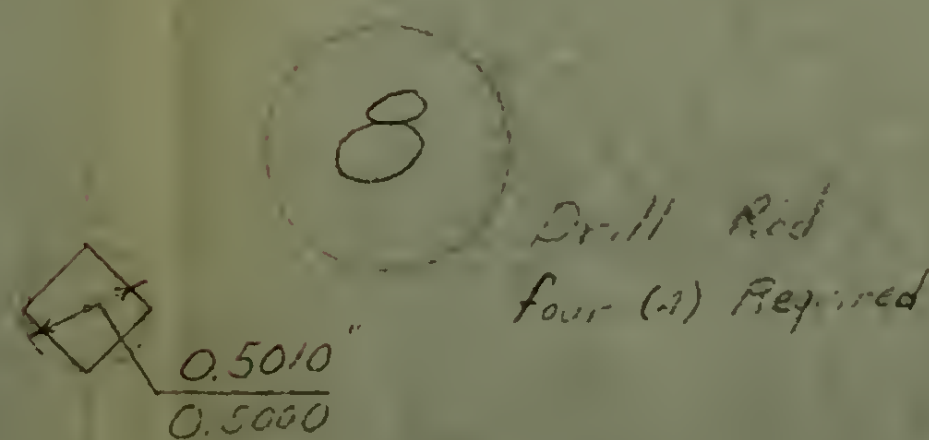
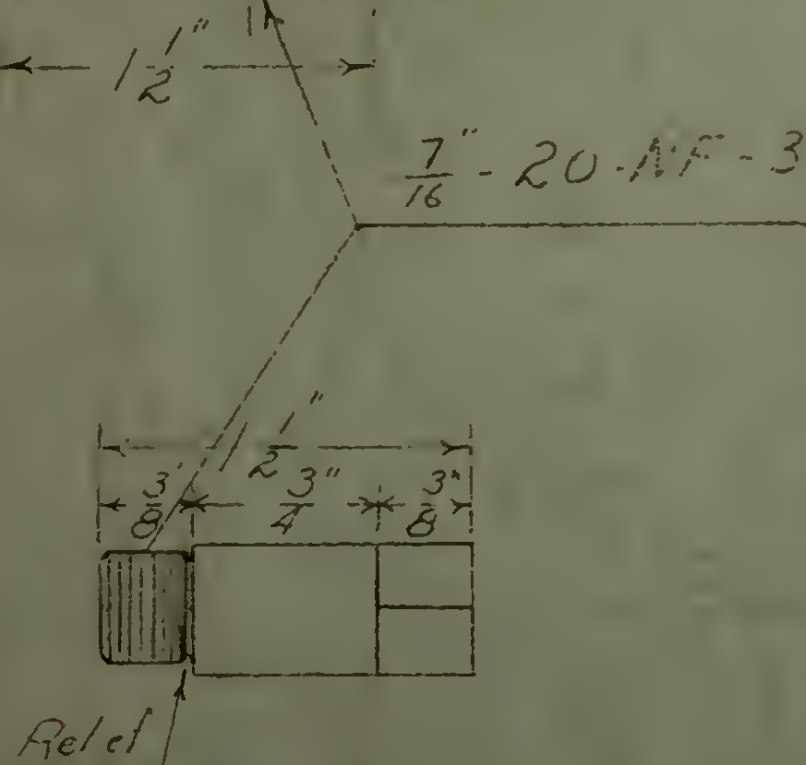
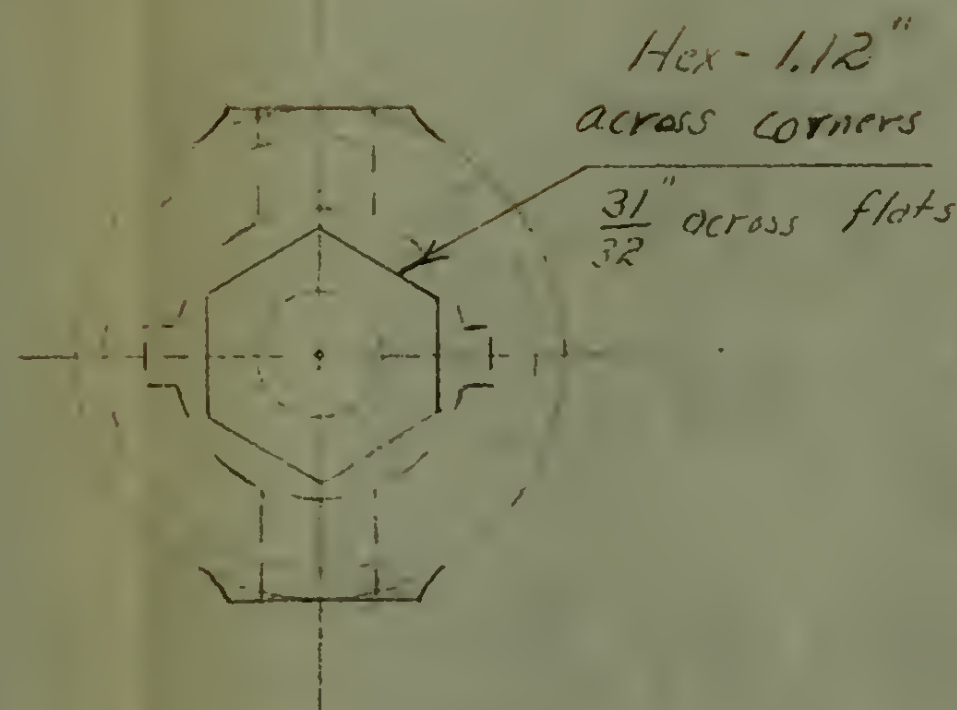
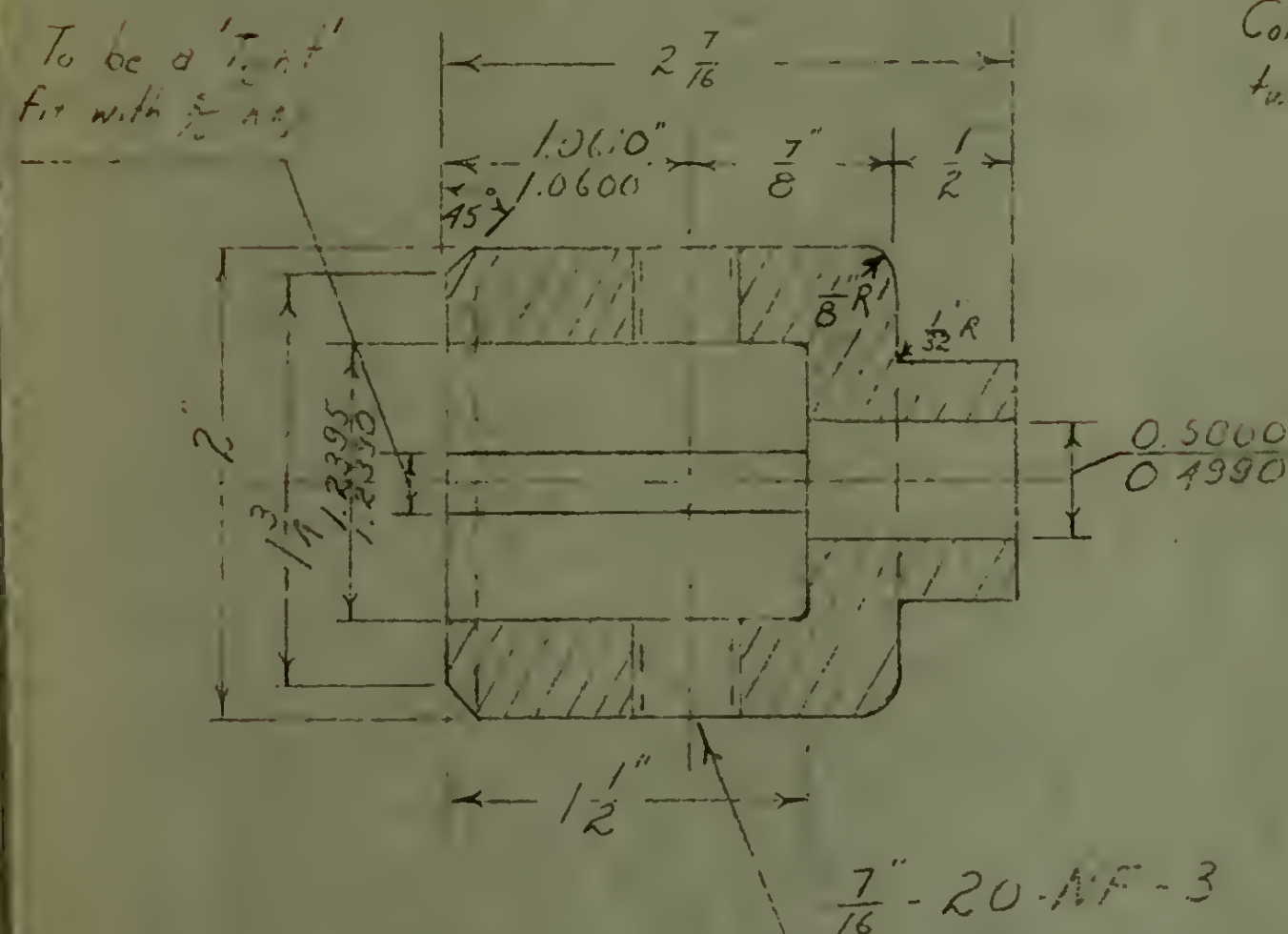






7

Cold Rolled Steel  
Two (2) Required



Notes

Material ~ As indicated

All tolerances  $\pm 0.0020$  except as specified

Fine machine finish throughout

U.S. Naval Postgraduate School  
Monterey, California

Combined Stress Fatigue Test

Specimen Holder Gage

A.J. Gilmore

1 March 1953



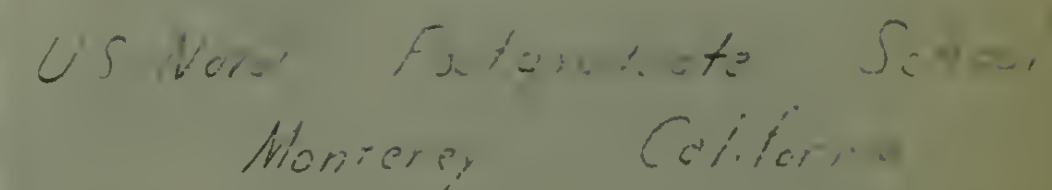


Meter, u ~ 615T-6

All tolerances  $\pm 0.02$  in, unless otherwise noted.

Fire machine finish through

Two (2) required



Combined Static Fatigue Test

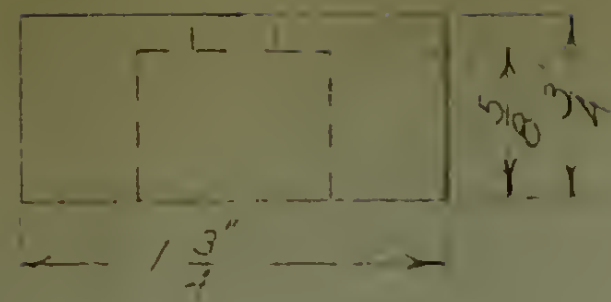
For. Fern

AJG:more

3 May 1952







16

6157-6  
Four (4) required

Notes

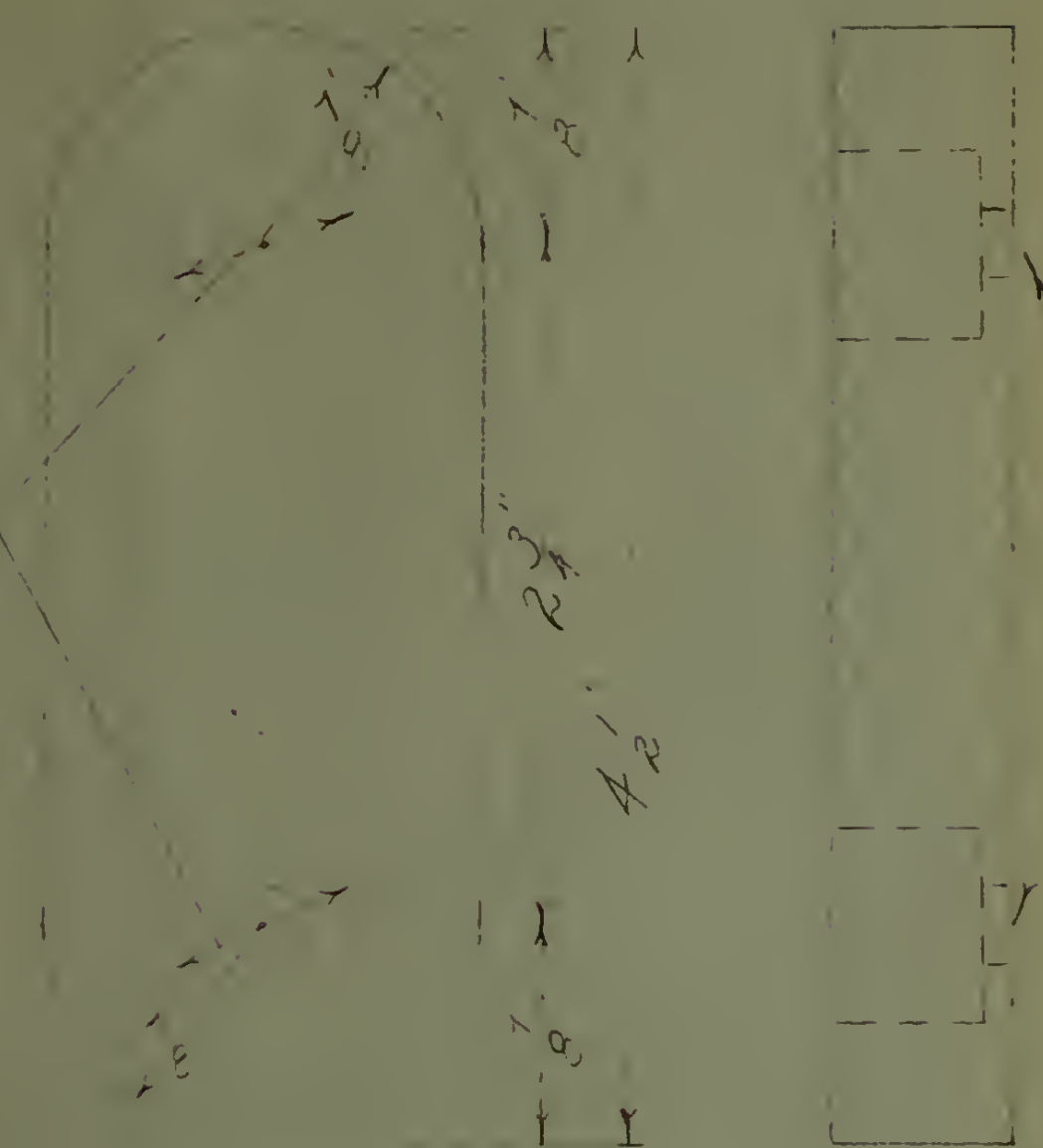
Material - 1/2 inch plate

All tolerances to be as shown on drawing

Five machine screws in all

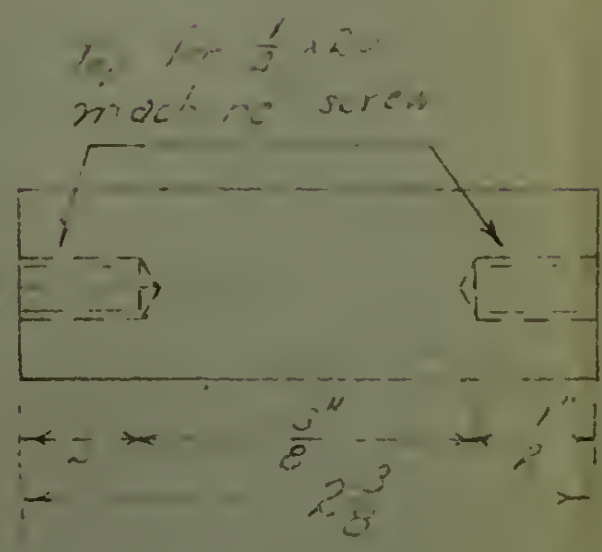
Four (4) each required

To be a Tight  
fit with part 15



1/4 inch  
machine screw

To be a Press fit with  
Tolerance .0012-.0015  
part 14



15

Gal. 3/16 Steel  
Four (4) required

U.S. Naval Postgraduate School  
Monterey, California

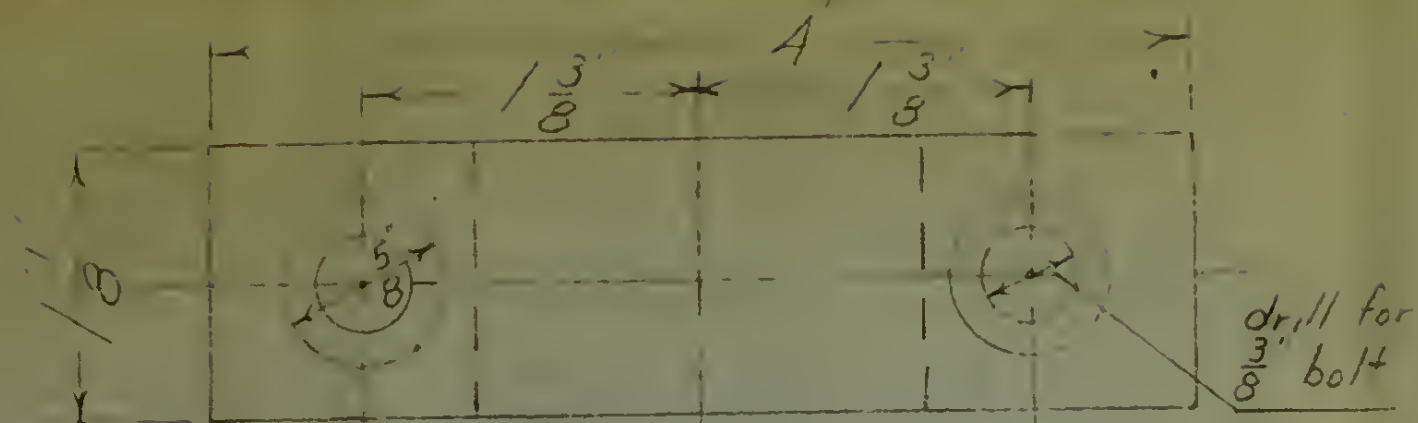
Combined Stress Fatigue Test

For Stress in the joint

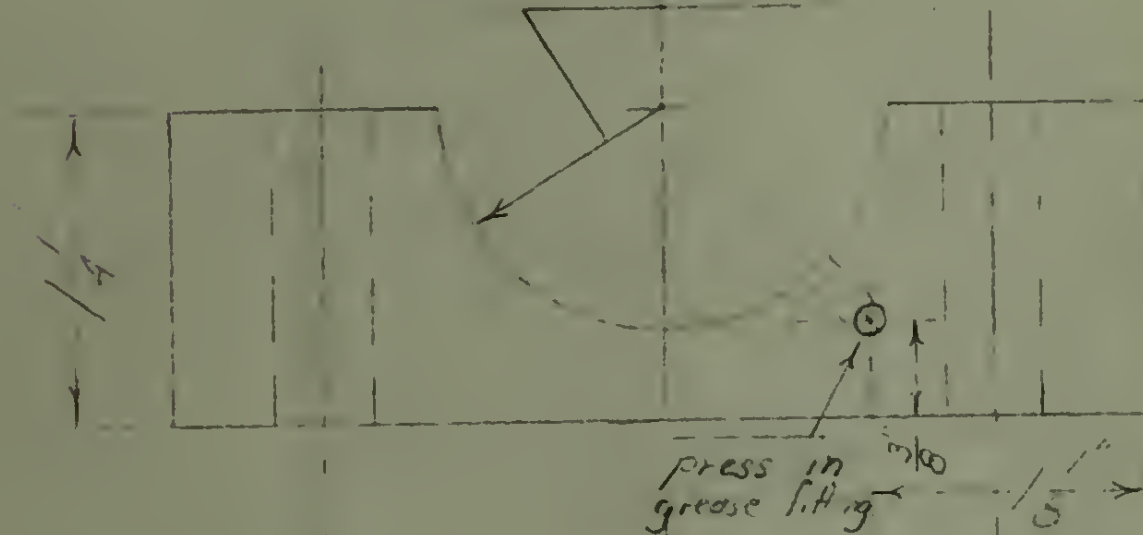
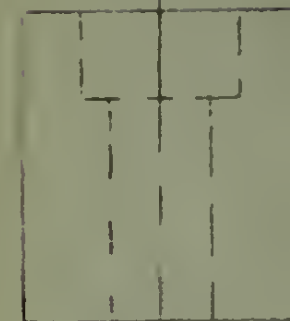
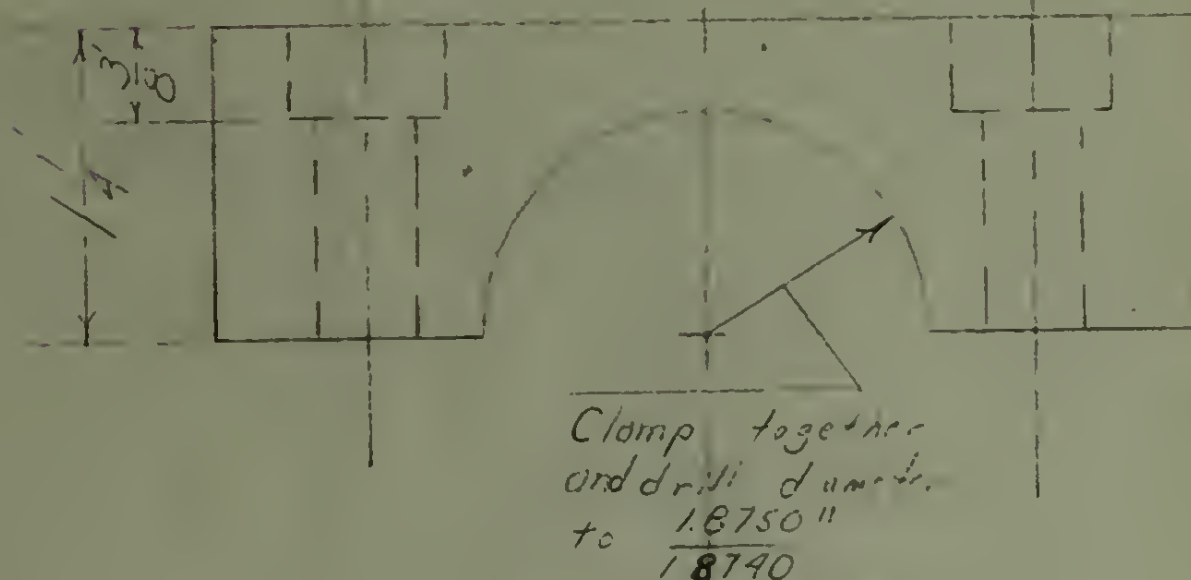
AJ Simmons

3 March 1952

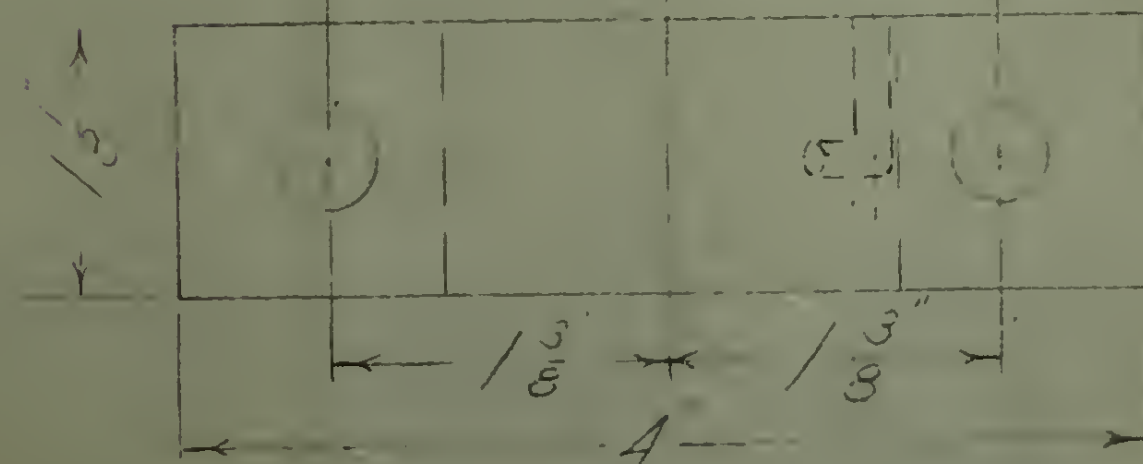
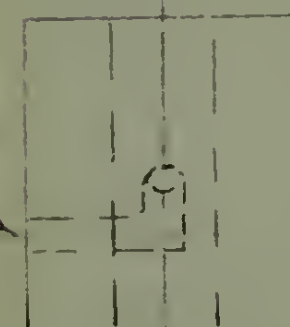




18



1/8" drill



17

## Notes

Material ~ Cold Rolled Steel

All tolerances  $\pm 0.0020$  except as specified

Fine machine finish throughout

Two (2) each required

U.S. Naval Postgraduate School  
Monterey, California

Combined Stress Fatigue Tug

Lower bearing holder and caps

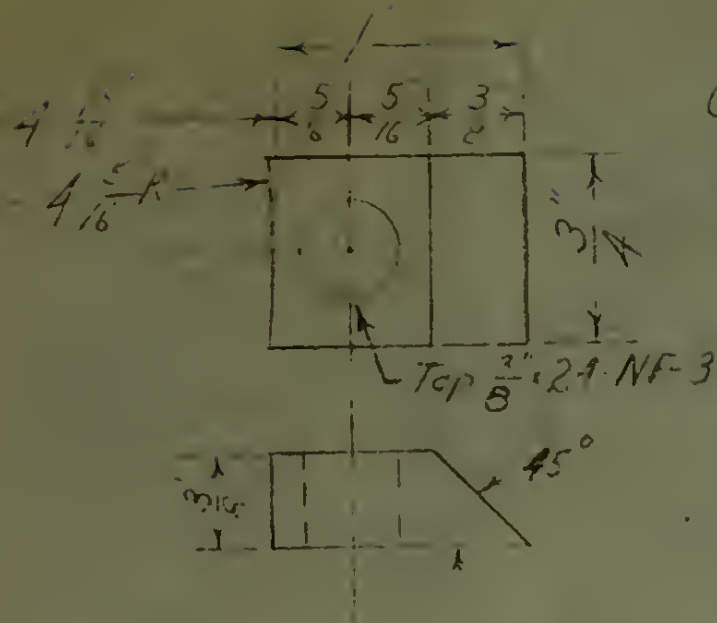
AJ Gilmore

4 March 1953

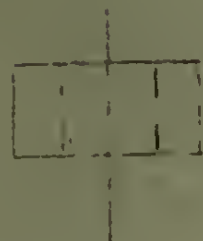




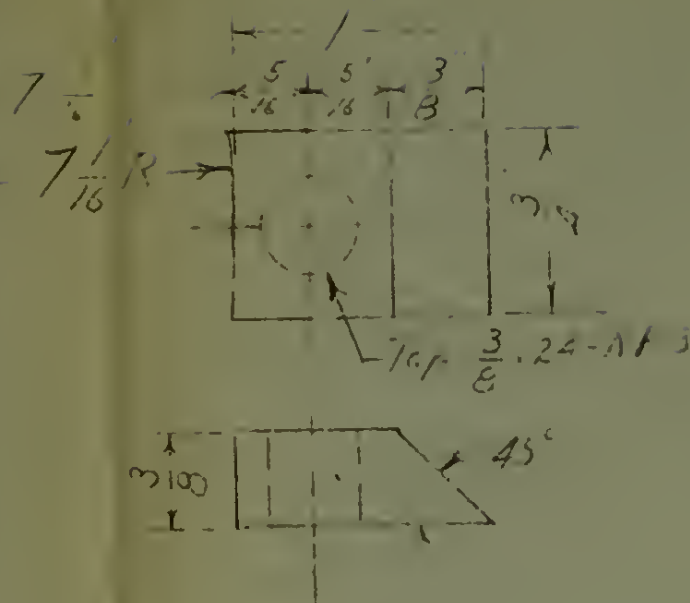
19a



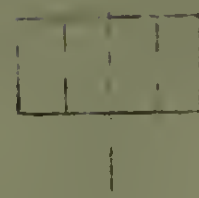
Cold Rolled Steel  
Two (2) Pieces



19b



Cold Rolled Steel  
Two (2) Pieces



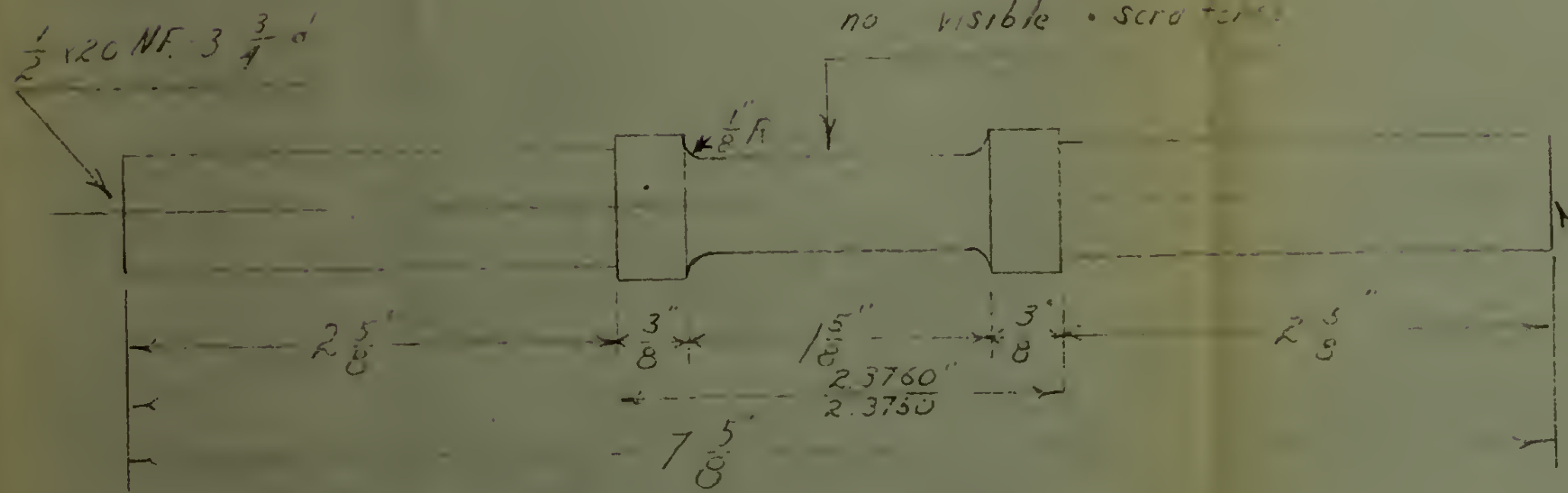
Notes

Material ~ As specified

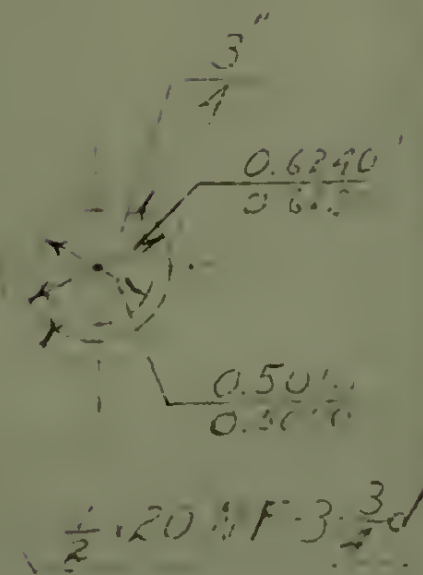
All tolerances  $\pm 0.0020$ " except as specified

Fine machine finish throughout

Polish to mirror finish  
no visible scratches



Specimen



U.S. Naval Postgraduate School  
Monterey, California

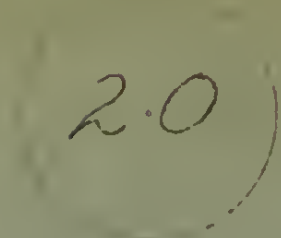
Combined Stress Fatigue Test

Tribe clip and Specimen

H.J. Gilmore

5 March 1953





Notes



All tolerances  $\pm 0.0020$  except as specified

Fine machine finish throughout

Two (2) required

Half Scale

U.S. Naval Postgraduate School  
Monterey, California

Combined Stress Fatigue T<sub>19</sub>

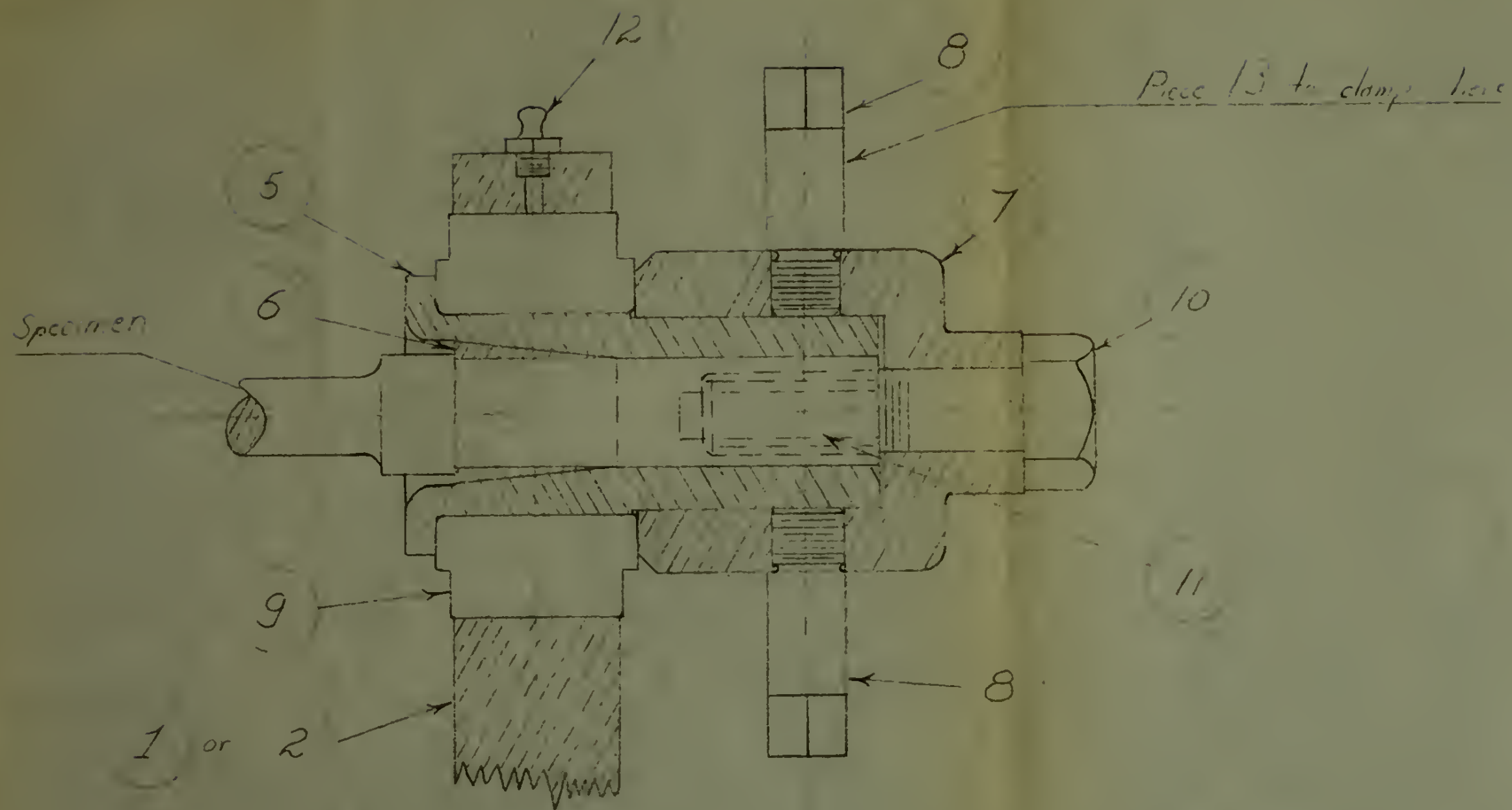
table	top
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A. J. Gilmore

9 March 1955







U.S. Naval Postgraduate School  
Monterey, California

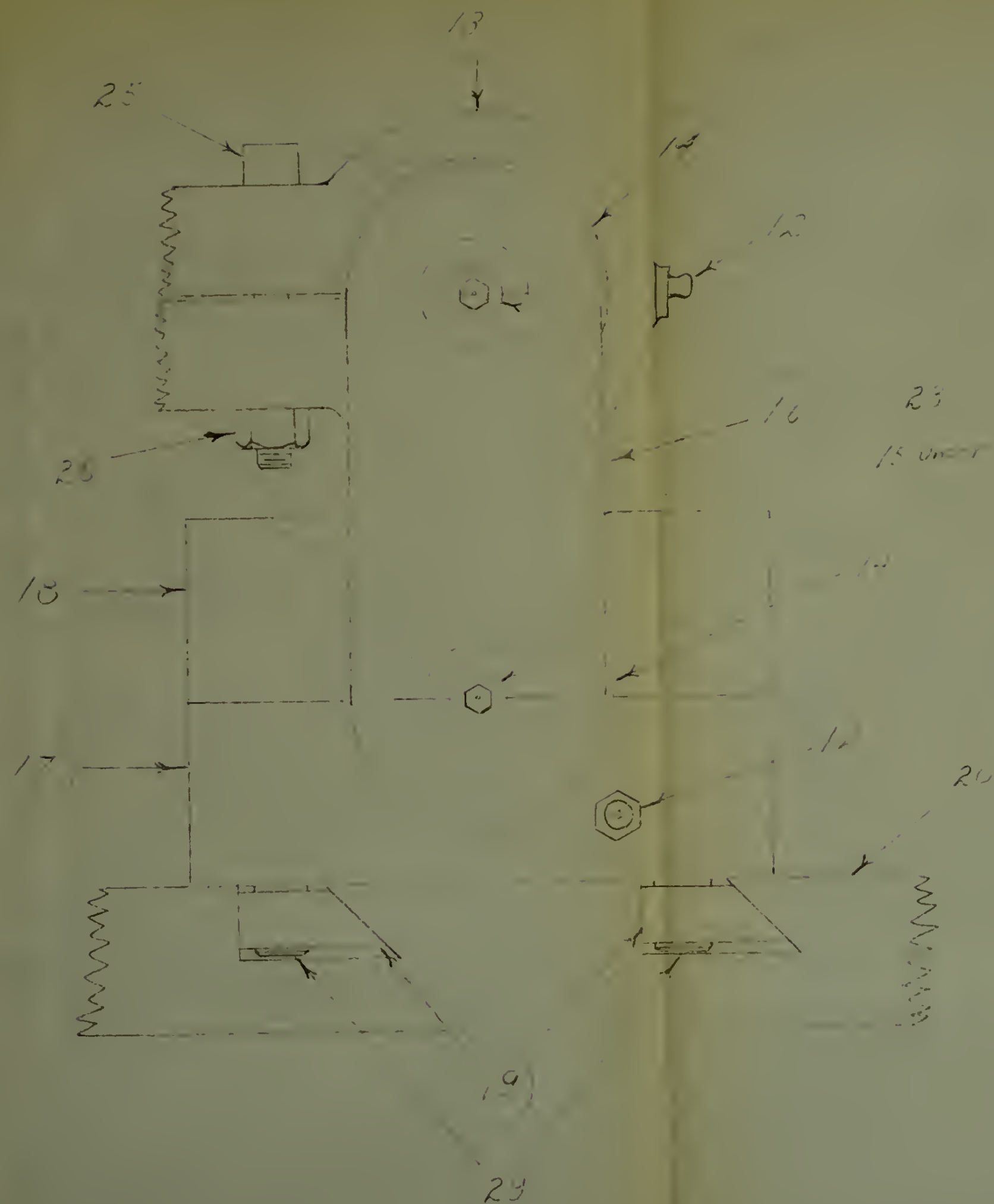
Combined Stress Fatigue Jig

Assembly of specimen holder

AJ. Gilmore

3 March 1961





U.S. Navy Postgraduate School  
 Monterey, California

Combined stress fatigue test  
 Assembly of hold down

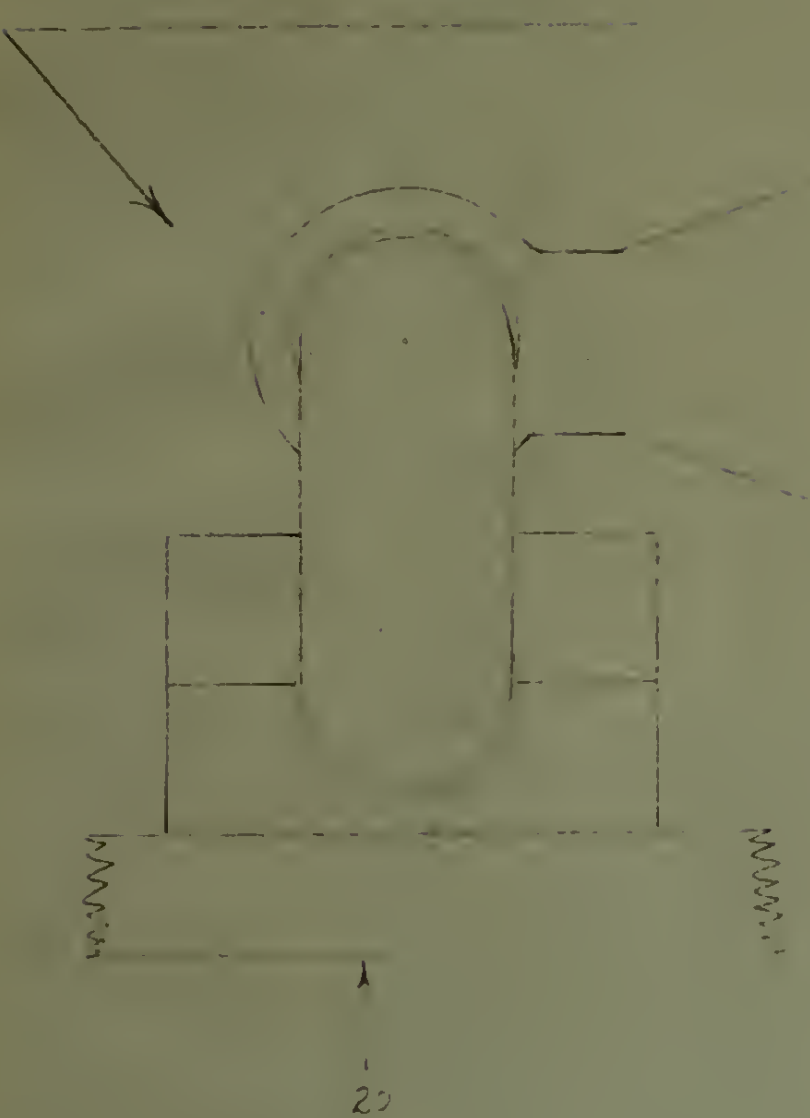
H.J. Gilmore

3 March 1940

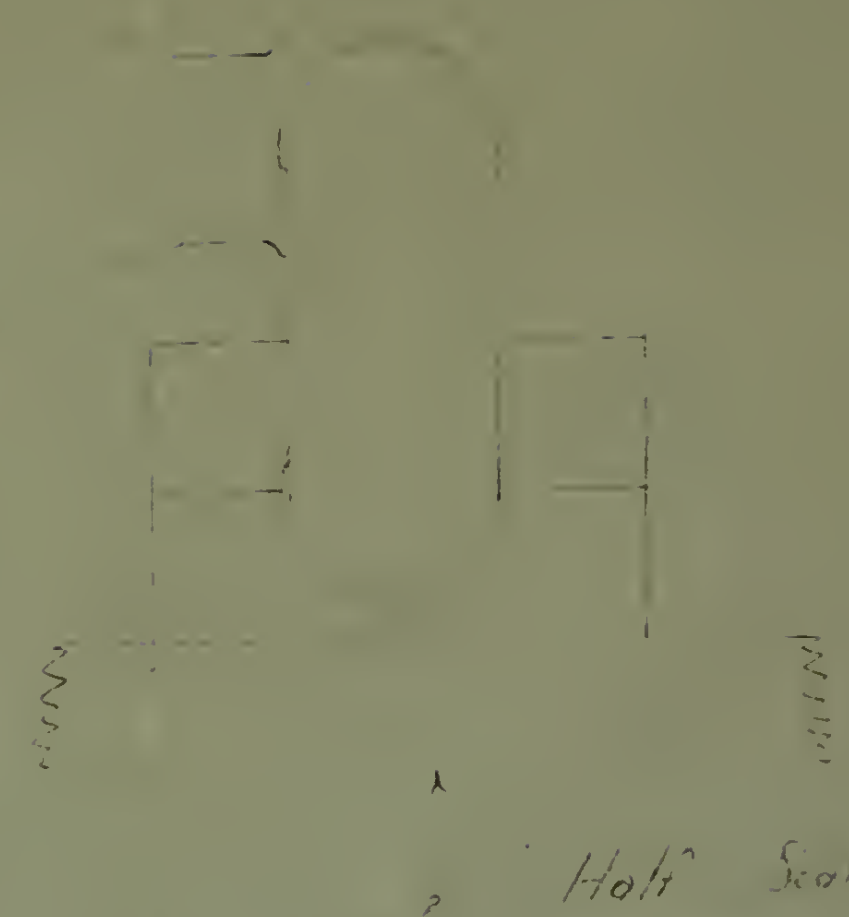
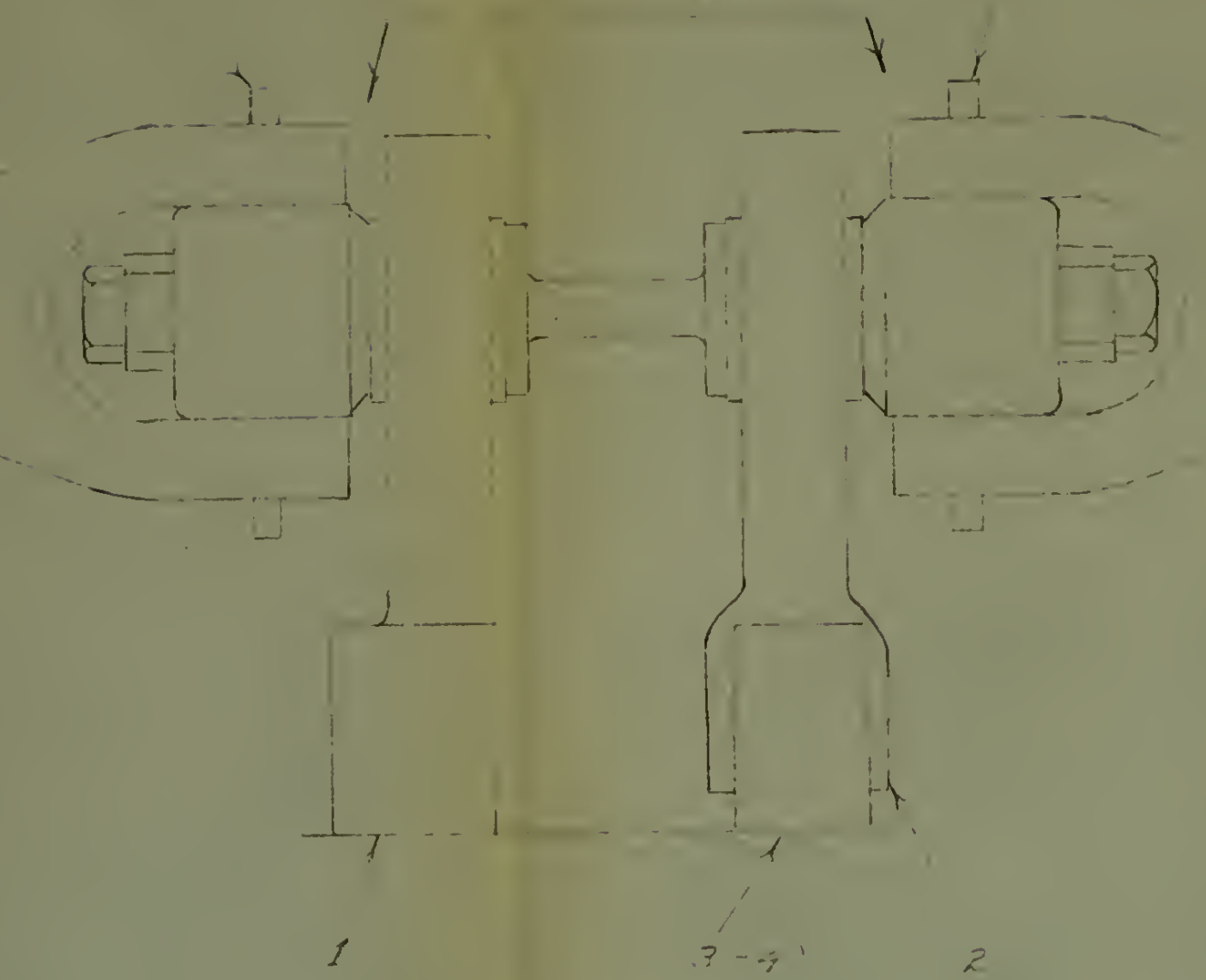




See sub-assembly of  
hold - turn section



See sub-assembly of  
symmetrical holding section



Half Scale

U.S. Navy Designation: SSM /  
Winterer: C. J. C. C.

Continued: [unclear] [unclear]

B. J. L. [unclear]

March 1944













JUL 2  
FEB 8  
MAR 11  
APR 23  
MAY 11  
JUN 10

BINDERY  
889  
RENEWED  
401  
RENEWED  
306  
306  
1377

Thesis Gilmore 20657  
G452 The design of a combined  
stress fatigue jig.

FEB 8  
MAR 11  
APR 23  
MAY 11  
JUN 10  
AG 3062

BINDERY  
889  
RENEWED  
401  
RENEWED  
306  
11377

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fatigue jig.

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Monterey, California





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The design of a combined stress fatigue



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